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AVADANLIQLAR, TEXNOLOGİYALAR, MATERİALLAR

ОБОРУДОВАНИЕ, ТЕХНОЛОГИИ, МАТЕРИАЛЫ

VOLUME 11 ISSUE 03 2022

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MÜNDƏRİCAT / TABLE OF CONTENTS

EQUIPMENT. TECHNOLOGIES. MATERIALS

VOLUME 11 ISSUE 03 2022

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CAVITATION IN OIL IMPELLER PUMPS

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ABSTRACT

Analysis shows that many liquids contain soluble gas in them. Such fluids have low viscosity and density. However, when pumping liquids, a number of problems arise, one of which is the appearance of bubbles in the mixture. Such liquids with gas do not obey the law of homogeneous liquid and gas. Therefore, in recent years, special attention has been paid to the hydromechanics of two-phase systems, such as oil-gas, water-steam, etc. the appearance of gas bubbles also affects the operation of centrifugal pumps, which manifests itself during pumping.
Keywords: centrifugal pump, gas-liquid system, gas bubbles, torque on the shaft, power, mass of liquid.

Introduction. Recently, the bulk of oil is pumped by centrifugal pumps. Many years of pumping experience shows that many extracted oils are liquids with saturated gas. It should be noted that at the last stage of operation of the largest deposits, hydrogen sulfide gas appears in significant quantities in the production of wells. Such a manifestation worsens the quality of oil, complicates their pumping and processing, creates difficulties during transportation. Hydrogen sulfide is also dangerous in terms of corrosion. Pipelines and pumps are destroyed from corrosion, the tightness of these objects is destroyed in branched pipelines. As a result of this case, emergency multi-ton spills of oil and mixtures into the environment and its irreversible pollution occur. Therefore, corrosion protection of all equipment, pump, pipeline is an important technical, economic and environmental task. [1-5].

Note that reliable corrosion protection is the accounting of gas particles that occur when pumping liquid using a centrifugal pump. One of the main requirements for pumps is their functionality, they must be able to simultaneously not only protect against various types of accidents, but also slow down the appearance of gas bubbles, i.e. the amount of hydrogen sulfide in the product. The purpose of this work is to study the effect of gas bubbles on the operation of a centrifugal pump and pipeline.

Research methods. Practice shows that centrifugal pumps are the most widely used of the vane pumps. These pumps are used in almost all areas of the oil industry - water supply, construction, oil transportation through pipelines, and especially often in the chemical and petrochemical industries. Note that in this case, additional energy makes it possible to increase the initial pressure.

Solving the problem: It is known that the transfer of energy to the fluid flow from the electric motor is carried out by means of an impeller. When the impeller rotates, the gas-liquid mixture filling the space between the blades comes into rotation. Under the influence of the forces developing at the same time, the mixture is pumped to the periphery of the wheel. At the same time, the circumferential velocity increases. Under the action of pressure transfer, the mixture rises through the suction pipeline. The increment of the energy of the mixture in the impeller occurs mainly due to an increase in the circumferential speed.
To solve this problem, we assumed that the mixture moves between the blades of the impeller in parallel streams, there is no vortex motion and there is no pressure loss when the mixture moves through the channels.
Given the well-known theorem in mechanics about the change in the moment of the amount of motion, we can write:

$$M_2 c_2 l_2 - M_1 c_1 l_1 = M_{кр} \Delta t \quad (1)$$

where M_1 *and* M_2 – the mass of liquid that has passed through the impeller during the time from start to finish;
l – the shoulder of the moment of the amount of movement;

$$m = \rho Q \Delta t \quad (2)$$

$$l_2 = r_2 cos\alpha_2 \text{ и } l_1 = r_1 cos\alpha_1 \quad (3)$$

C_1 *and* C_2 – absolute velocity vector;
α_1 *and* α_2 – vanishing angles of the mixture.
Since we are dealing with a mixture, the mass of the mixture consists of:

$$m = m_g + m_l \quad (4)$$

Then , taking into account the density and volume of the liquid and gas , it can be written as:

$$\rho = \rho_g \frac{V_g}{V} + \rho_l \frac{V_l}{V} \quad (5)$$

where ρ_l и ρ_g – liquid and gas densities,
V_l *and* V_g – volumes of liquid and gas,
ρ – density of the gas-liquid mixture,
V – total volume of the gas-liquid mixture.
Solving with respect to time, we have:

$$\rho = \rho_g \frac{Q_g}{Q} + \rho_l \frac{Q_l}{Q} \quad (6)$$

where Q_l *and* Q_g – volumetric flow rate of liquid and gas.
Then we have:

$$m = \rho Q = \rho_g Q_g + \rho_l Q_l \quad (7)$$

On the other hand

$$M_{\text{кр}} = \frac{N_t}{\omega} = \frac{\rho g Q H}{\omega} \quad (8)$$

where N_t – theoretical power on the shaft of an impeller with an infinitely large number of blades,
ω – angular speed of rotation of the impeller,
H – pressure generated by the pump.
Solving equations (8) and (1) together we have

$$\frac{m_2 c_2 l_2 - m_1 c_1 l_1}{\Delta t} = \frac{\rho g Q H}{\omega} \quad (9)$$

The operation of a centrifugal pump is characterized by its supply, pressure, power consumption, efficiency and rotation frequency. It is known that the supply of the pump is called the flow of liquid through the pressure pipe. Just like the flow rate, the feed can be volumetric and mass. Note that the pressure is the energy difference of the unit weight of the liquid in the pump section after the pump.

$$H = Z_1 - Z_2 + \frac{P_1 - P_2}{\rho g} + \frac{v_1^2 - v_2^2}{2g} + h_{\text{т}} \quad (10)$$

where H – full pressure generated by the pump,
Z_1 and Z_2 - geometric pressure at the beginning and at the end of the installation,
P_1 and P_2 – pressure at the beginning and end of the pump,
v_1 and v_2 – flow rate at the beginning and end of the pump,
ρ – liquid density,
g – acceleration of free fall,
h_f – friction pressure loss.
If the system consists of two phases, i.e. gas and liquid, then the density is defined as:

$$\rho = \rho_l (1 - \varphi) + \rho_g \varphi \quad (11)$$

where ρ_l and ρ_g – liquid and gas densities, φ – true gas content.
The analysis showed that the true gas content can be determined from the volume gas content:

$$\varphi = 0{,}833 \cdot \beta \quad (12)$$

where β – volumetric gas content.

$$\beta = \frac{Q_g}{Q_g + Q_l} \quad (13)$$

where Q_g and Q_l – volumetric flow rate of gas and liquid.
Given that there is gas in the system, we have:

$$\Delta P_l = \frac{P}{(1 - \varphi)^{1.5}} \quad (14)$$

where ΔP – pressure losses in the gas-liquid system mixture,

ΔP_l – pressure loss for a homogeneous liquid.
As can be seen, the occurrence of gas in the liquid flow affects the operation of the pump, i.e., depending on the volumetric gas content, the total pressure of the created pump varies.
In this case, the pump power also changes:

$$N = QgH\rho \tag{15}$$

where Q – total consumption of the mixture.
Thus, we have obtained the equations of the pressure and power of the centrifugal pump, taking into account the flow rate of the gas in the mixture pumping through the pump.

Conclusions: 1. The basic equations of the centrifugal pump are obtained taking into account the gas in the mixture.
2. The gas in the mixture changes the pump parameters, which must be taken into account when designing and constructing pipelines and the pump.
3. The increase or decrease in pump parameters depends on the gas flow in the systems, which is an urgent problem in the design of pipelines with centrifugal pumps.

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КАВИТАЦИЯ В НЕФТЯНЫХ ЛОПАСТНЫХ НАСОСАХ

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РЕЗЮМЕ

Анализ показывает, что многие жидкости в себе содержат растворимый газ. Такие жидкости обладают малой вязкостью и плотностью. Однако, при перекачке жидкостей возникает ряд проблем, одним из которых является возникновение пузырьков в составе смеси. Такие жидкости с газом не подчиняются законом однородной жидкости и газа.

Поэтому особое внимание в последние годы обращают на гидромеханику двухфазных систем, типа нефть-газ, вода-пар и т.д. появление газовых пузырей влияет и на работу центробежных насосов, что проявляется при перекачке.
Ключевые слова: центробежный насос, газо-жидкостная система, пузырьки газа, крутящий момент на валу, мощность, масса жидкости.

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XÜLASƏ

Təhlil göstərir ki, bir çox mayenin tərkibində həll olunan qaz var. Belə mayelər aşağı özlülük və sıxlığa malikdir. Bununla belə, mayelərin vurulması zamanı bir sıra problemlər yaranır, bunlardan biri maye qarışıqında qabarcıqları görünüşüdür. Belə qazlı mayeler homoqen maye və qaz qanununa tabe olmur. Ona görə də son illərdə ikifazalı sistemlərin, məsələn, neft-qaz, su-buxar və s., qaz qabarcıqlarının görünüşü də mərkəzdənqaçma nasoslarının işinə təsir göstərir ki, bu da nasosun çəkilməsi zamanı özünü göstərir.
Açar sözlər: mərkəzdənqaçma nasosu, qaz-maye sistemi, qaz qabarcıqları, valdakı fırlanma momenti, güc, mayenin kütləsi.

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ANALYSIS OF THE CURRENT STATE OF THE USE OF POLYMER MATERIALS IN THE STRUCTURES OF OIL AND GAS EQUIPMENT

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ABSTRACT

The article discusses the possibilities of using polymer materials and compositions based on them, as well as nanotechnology in the manufacture of parts and assemblies included in the design of oil and gas field equipment, in order to increase their reliability and durability.

Keywords: oil and gas field equipment, nanotechnology, reliability.

Modernization of machinery and equipment applied in the oil sector, improvement of their quality indicators and reliability are considered as priority issues. Improvement of reliability, longevity and efficiency of oil-drilling equipment ODE should be given special attention to replacing traditional non-ferrous and non-ferrous metals with new composite materials (including polymer-based) applied in existing constructions on the basis of innovative technologies or covering the working surface of them aasem 1,2.

Since the middle of the XX century, the use of plastic materials in the preparation of parts and nodes of ODE, as well as equipment used in oil and gas processing processes began. In the following years, the application of polymeric composition and nanomarerials began to gain momentum.

In the second half of the XX century, if more than 300 parts for different purposes were made of wide-spectrum plastic mass [3], this level in modern oil-field equipment decreased to 120. These can be explained by objective and subjective factors. Thus, over the past 20 years, the decline in the oil engineering sector in the country, the change in the production profile of some enterprises and the low competitiveness of the ongoing production of products have had a significant impact on the range and number of the ODE.

Table 1 shows the most active period of plastic materials in the production of ODE in the Republic of Azerbaijan compared to 1970, last 20 years in this area.

As can be seen from the analysis of Table 1.1, when the use of plastic materials applied in parts and nodes of the ODE significantly decreased compared to 1970, the number of them began to increase in dynamics in the following years.

Table 1. Information about polymeric materials applied in oilfield equipment

Indicators	1970	By years					
		1995	2000	2005	2010	2015	2020
Number of parts made of polymer material, PCs	300	52	75	78	90	91	98
Thermoplasts, ton	258,3	46,6	65,5	67,2	81,5	85,8	86,6

Reactoplasts, ton	85,2	16,8	22,1	34,1	45,9	57,6	66,2
Coating materials, ton	2123,5	1224,3	1625,8	1764,3	1805,5	1866,4	1902,2

Depending on the operating conditions and constructive characteristics of the oil drilling equipment, various requirements are imposed on the plastic materials applied here. The requirements of machine parts made of metal are classified in 4 groups depending on their functional purpose: connecting, sealing, parts used in power and friction units. As a result of our research, it has been established that the requirements in parts made of polymeric materials can be grouped according to the following indicators:
-by designation: threaded parts, transmission nodes (rotation, displacement, etc.);
- according to geometric characteristics;
-according to its constructive structure: monolithic and hybrid parts, fittings and parts included in the plug Assembly.
- polymer-based coating materials.
Taking into account the above, polymeric materials used in the structures of the ODE can be included in the following requirements: high mechanical strength, dimensional accuracy, wear resistance and resistance to external aggressive environment. In addition, it should be borne in mind that in the country's conditions, oil and field equipment is affected by a wide range of atmospheric changes, and it is very important to take into account these factors. The fact that the operating conditions are below and above the norm leads to the rapid disintegration of the materials used in the structures of the ODE or to a change in their properties. Also, due to air pollution, intensive wear of rubbing surfaces of parts can occur, for example, in Absheron Peninsula, there are frequent winds accompanied by storm, as a result of which large amounts of dust and sand are emitted into the air. It has been established that the climatic conditions in yistism are very complex, sometimes the atmospheric temperature varies within the limits from -20 PPPs to + 48 PPPs.
Depending on the seasonal changes, the average annual relative humidity in the country is 70-72%, in winter it rises to 81-88%, and in summer it falls to 50-55% [4].
One of the factors affecting details for oilfield equipment is their high abrasive (for example, washing solutions used in drilling) and aggressiveness (processing of wells with salt-acid liquids). From the point of view of constructive features, oilfield equipment can be characterized by the mobility of its components, the integrity of the compounds and The Shape of the surface of the details.
In recent years, polymeric materials in the practice of oil drilling have been used independently or in composition for corrosion protection of pipelines, large bodies, tanks and various devices [5]. Due to the fact that most of these equipment and facilities pass under the ground, as well as being in contact with aggressive environment, protection of their working surfaces from corrosion is topical. There are a lot of methods for this purpose, and the use of polymer coating materials dominates.
According to the list determined on the basis of analysis of the structures of parts and nodes of the existing oil-field equipment, the use of polymeric materials in them allows to increase the economic efficiency, as well as their longevity confirm the replacement of metallic materials in these structures with polymers or composites made on their basis.

Details prepared at ASOIU as a result of scientific-research works carried out and currently carried out in the direction of replacing the metal parts of the ODE with polymeric materials Figure 1.- also reflected.

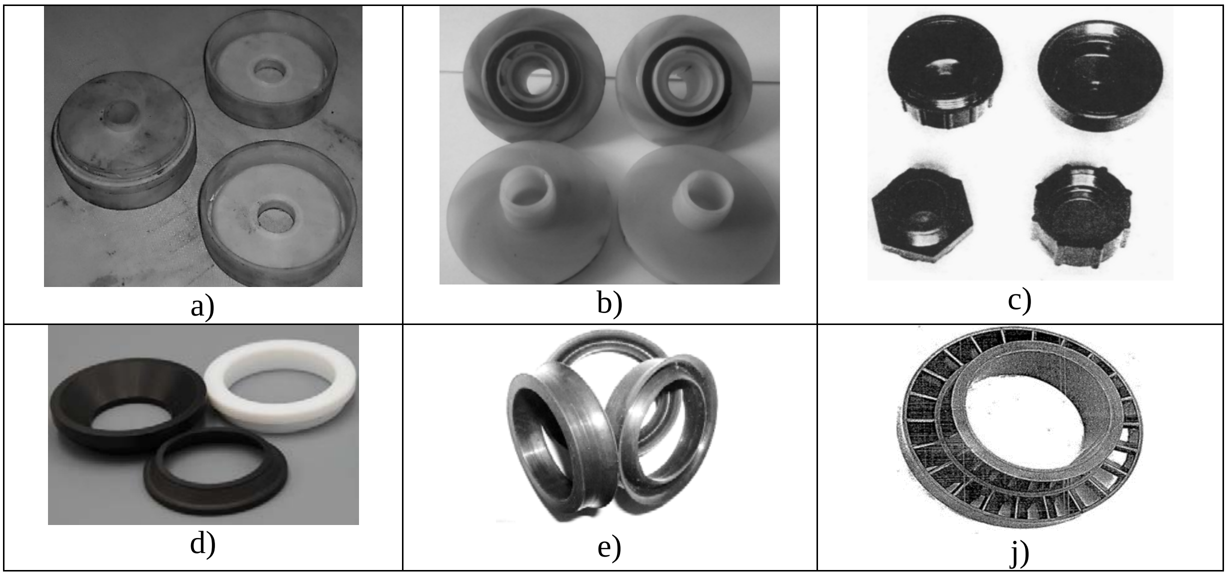


Figure 1. Parts and knots of oil equipment made of plastic and polymer based composition materials. a,b – working step of submersible pump; d – threaded caps made of thermoreactive materials; d - sealing cuffs; e–drilling pump polymer - based composites; J-rotor of truboburus from prestoz.

As can be seen from the analysis of Figure 1, these details are applied mainly to protect the grooved surfaces from damage, which are not subject to force and overloads, as well as to protect them from damage. The requirement for obtaining accuracy is put as a key factor in their preparation. However, in some cases, it is required to solve complex requirements for the preparation of precision abrasive wear resistance (filter pumps), as well as parts subject to friction wear (working rod of electric submersible Nano).
Since 2015, the use of nanotechnology has been expanding among the innovative measures taken to increase the longevity and effectiveness of ODE [6, 7].
As a result of the research, it has been established that, using nanotechnology, it is possible to significantly increase the leading characteristics and life of plastic materials, as well as polymer coatings.
Figure in 2, hybrid (metal+polymerized composite) parts used in the structures of equipment from oil were reflected.



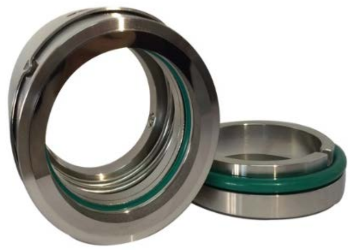
a)	b)	c)
 d)	 e)	 j)

Figure 2. Hybrid (composite based on metal+polymer) parts used in oil-bearing equipment: a, b-piston of oil-bearing pumps; c-metal + polymer seals; d-valve of oil-bearing pump; e-spring elastamer rings used in water pumps.

Results.

-as a result of analysis of operational environment of oil-field equipment and prospects of application of polymeric materials in them, it was established that if application of polymeric materials in NMA structures decreases, its dynamics develops on an upward line.

- in this sector of the industry, polymeric materials and composite materials prepared on their basis are different from the parts used in general machine construction.

- the development direction of the use of polymeric materials in the structures of oil drilling equipment is related to the level of application of nanotechnology here.

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NEFTQAZMƏDƏN AVADALIQLARININ QURULUŞLARINDA POLİMER MATERİALLARDAN İSTİFADİNİN MÜASİR VƏZİYYƏTİNİN TƏHLİLİ

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XÜLASƏ

Məqalədə polimer materialların və onların əsasında hazırlanan kompozisiyaların neft-qaz-mədən avadanlığının konstruksiyalarına daxil olan detalların və qovşaqların etibarlılığının və uzunömürlülüyünün artırılması məqsədi ilə eləcə də hazırlanmasında nanotexnologiyalar, təbiqi məsələləri nəzərdən keçirilir.
Açar sözlər. neft-qaz avadanlığı, nanotexnologiya, etibarlılıq**.**

АНАЛИЗ СОВРЕМЕННОГО СОСТОЯНИЯ ИСПОЛЬЗОВАНИЯ ПОЛИМЕРНЫХ МАТЕРИАЛОВ В КОНСТРУКЦИЯХ НЕФТЕГАЗОВОГО ОБОРУДОВАНИЯ

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АННОТАЦИЯ

В статье рассматривается возможности применения полимерных материалов и композиции на их основе, а также нанотехнологии при изготовлении деталей и узлов, входящие в конструкции нефтегазопромыслового оборудования, с целью повышения их надежности и долговечности.
Ключевое слова: Нефтегазопромысловое оборудования, нанотехнология, надежность.

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RESEARCH ON THE RESERVES FOR REDUCING THE METAL INTENSITY OF COMMERCIALLY AVAILABLE DOWNHOLE SUCKER-ROD PUMPING UNITS

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ABSTRACT

A modern well pumping unit (WPU) consists of downhole equipment, including a downhole sucker-rod pump (BHP), tubing string and pump rods, and surface equipment, including the drive of the downhole sucker-rod pump and wellhead equipment. Each of these elements is serially produced by oil industry in accordance with state standards. In the given article with the purpose of finding out the possibility to decrease metal consumption of serial rocking units the possibility of transition to one-beam welded design of the rocking unit balancer is investigated.

Keywords: rocking-type machine, rocker, single-beam design, metal-intensity,

Significance of the study: At the present time in the machine-building industry, metal saving is a priority. Given the large dimensions of the object under study: rocking machines, among the options to reduce metal intensity, we can consider the following works. For example: Transition to a beltless transmission in the transmission of the rocking machine. Transition to a single-beam welded design of the rocking machine balancer. Transition to a rocking machine gearbox design with full chevron meshing. Transition to a rocking machine design with modified kinematic ratios. In our work we turn to the next point, namely the transition to a single-beam welded rocking machine balancer design. In doing so, we replace the two-beam balancer design with a single-beam equal-strength design. By performing various design calculations, we will come up with metal savings.

Introduction: A modern well pumping unit (WPU) consists of downhole equipment, including a downhole sucker-rod pump (BHP), tubing string (tubing), pump rods, and surface equipment, including a downhole sucker-rod pump drive and wellhead equipment. Each of these elements is serially produced by the oil industry in accordance with state standards. Particularly, SSH drives are serially produced machines of SKD type of OST 26-16-08-87.
As a result of analysis of the manufactured rocking machines designs and methods of their calculation the following possibilities are suggested as measures, which can lead to sufficiently tangible metal saving:
-Transition to a beltless transmission in the transmission of the rocking machine.
-Transition to a single-beam welded design of the rocking machine balancer.
-Transition to a rocking machine gearbox design with full chevron meshing.
-Transition to the rocking machine design with modified kinematic ratios.

The development of the design of SSH drives with a beltless gear should be considered promising. In such a drive an angular reducer with bevel gearing at the high-speed stage and a multi-speed electric motor, allowing to change the rotation speed depending on the chosen operating mode, are used instead of a cylindrical reducer. The electric motor is connected to the reducer by an elastic coupling. Due to it the length of a rocking machine frame is reduced, there is no need for the use of a V-belt transmission with a set of massive pulleys, necessary for changing the number of shifts of a boom hanger point (BHP) of a rocking machine [1].
At the same time, the problem of refusal from using V-belts, the application of which causes difficulties constantly felt in the oilfield industry, is solved.
In this article, the possibility of transition to a single-beam design of a serial rocking machine balancer is considered.

Main part: The transition to a single-beam design allows you to save metal by eliminating the use of two I-beams under the condition of maintaining the same moment of resistance of the cross-section of the balancer. At the same time, the welded beam is somewhat higher than each of the two previously used beams, taking into account the fact that the bending moments acting on the balancer's body are not the same along its length. It is proposed to make a welded structure of variable height, which will also save metal.
Figure 1 shows cross-sections of a single-beam welded balancer (1 a) and a double-beam balancer of commercially available rocking machines rolled section (Fig. 1 b).

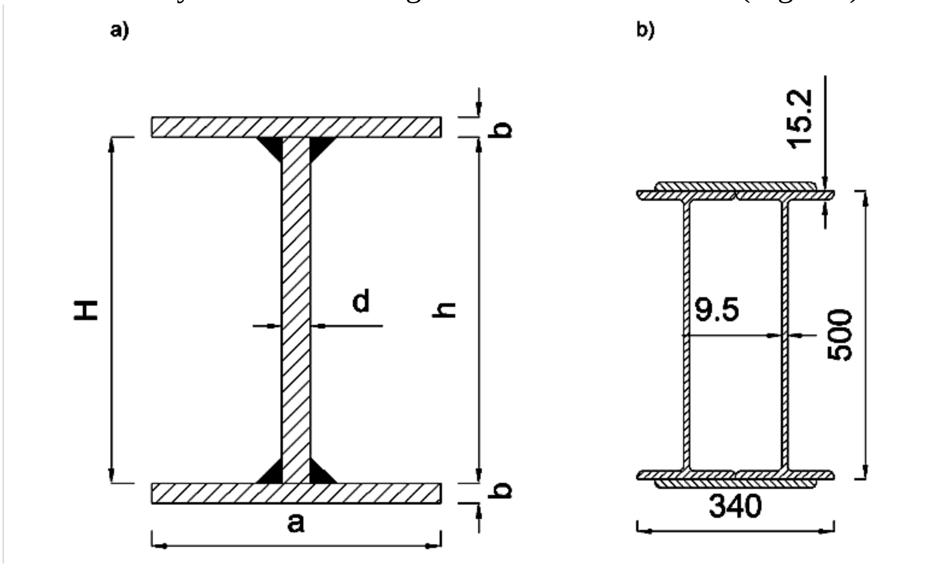


Figure 1. Schematic for the calculation of a single-beam welded balancer.

The bending stresses in the cross section of both balancers are determined by the formula:

$$\delta_{ben\,ser} = \frac{M_{ben\,ser}}{W_{ser}} = \frac{P_{max} \cdot K_1}{W_{ser}}$$

$$\delta_{ben\,wel} = \frac{M_{ben\,wel}}{W_{wel}} = \frac{P_{max} \cdot K_1}{W_{wel}}$$

where $M_{ben\,ser}$ and $M_{ben\,wel}$) are bending moments of the serial and welded balancer respectively; W_{ser} and W_{wel} are resistance moments of the serial and welded balancer respectively; P_max is the maximum load acting on the balancer head.

If we assume that the stresses in both sections must be the same, then their moments of resistance must be equal, since the length of the front arm k1 and the maximum load on the balancer head do not change.

The moment of resistance of the welded balancer cross-section is calculated by the formula (see Fig. 1a):

$$W_{ce} = \frac{a(h+2b)^3 - (a-d)h^3}{6(h+2b)} = \frac{a\left(1+2\frac{b}{h}\right)h^2 - (1-\frac{d}{a})h^2 a}{6(1+2\frac{b}{h})}$$

where h - height of the balancer wall; a - width of the balancer shelves; b - thickness of the balancer shelves; d - wall thickness of the balancer.

Denote

$$\lambda = \frac{b}{h};\ \lambda_1 = \frac{d}{a},$$

then

$$W_{wel} = \frac{ah^2}{6}\left[(1+2x)^2 - \frac{(1-\lambda_1)}{(1+2\lambda)}\right]$$

If we take a = 30 cm, h = 60 cm, b = 1.8 cm, d = 1.4 cm, then:

$$\lambda = 3 \cdot 10^{-2};\ \lambda_1 = 4{,}7 \cdot 10^{-2}$$

Substituting, we get:

$$W_{wel} = \frac{30 \cdot 60^2}{6}\left[(1+2\cdot 3\cdot 10^{-2}) - \frac{1-4{,}7\cdot 10^{-2}}{1+2\cdot 3\cdot 10^{-2}}\right] \approx 4140 \text{ cm}^3$$

If we compare the obtained result with the moment of resistance of the mass-produced and most frequently used rocking machine SKD8-3-4000, equal to W_{ser} =4649 cm^3 (by calculations of AZINMASH), we get:

$$\frac{W_{ser}}{W_{wel}} = \frac{4649}{4140} \approx 1{,}1.$$

I.e., it leads to an insignificant increase in bending stresses within the permissible limits. With some adjustment of the initial parameters of the welded balancer cross-section, it is possible to achieve full equality with the bending stresses of the serial balancer.

Let's calculate the weight of the welded balancer (Figure 2)

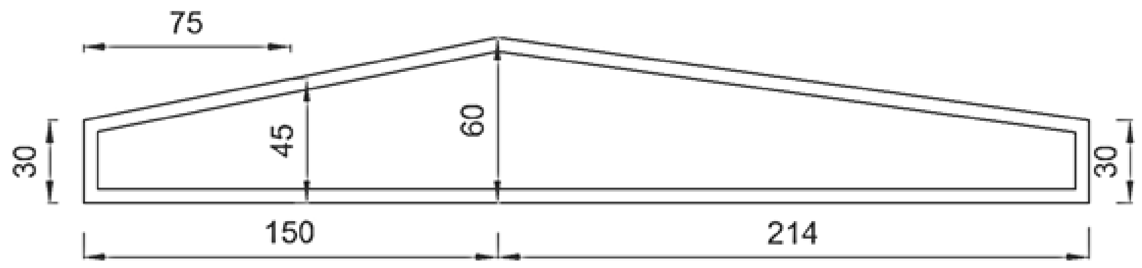


Figure 2. Diagram for calculating the weight of a welded balancer.

The length of the upper shelf of the welded beam l_{uf} will be calculated as follows:

$$l_{us} = \sqrt{150^2 + 30^2} + \sqrt{214^2 + 30^2} = 369{,}0\ cm$$

The weight of the bottom shelf l_{bs}=364 cm, the top shelf l_{us}=369 cm; the width of the shelf a=30 cm. The weights of the two shelves will be:

$$\sigma_{sh} = (364 \cdot 30 \cdot 1{,}8 + 369 \cdot 30 \cdot 1{,}8) \cdot 7{,}85 = 313{,}7\ kg$$

The weight of the wall will be:

$$\sigma_w = \left(30 \cdot 364 \cdot 1{,}4 + \frac{150 \cdot 30}{2} + \frac{214 \cdot 30}{2}\right) \cdot 7{,}85 = 150{,}3\ kg$$

The total weight of the welded balancer will be:

$$\sigma_{w.b.} = 313{,}7 + 150{,}3 = 463{,}0\ kg$$

Taking into account the weight of stiffeners, beams, weld etc., we take $G_{w.b.}$=470 kg.
Comparing the mass of the balancer body of the serial rocking machine SKD8-3-4000, equal to G_{ser}=514 kg, the metal saving was 44 kg.
The bending moment acting in the cross-sections of the body of a serial balancer varies from zero value at the balancer head to the maximum value in the section above the balancer support according to a linear law. Therefore, in case of a welded balancer design, it is possible to make it close to equal strength. For this purpose, the wall of the welded beam can be made smaller in height when approaching the balancer's head, keeping the height of the wall above the balancer's support taken as 600 cm and verified by calculation. Structurally, the height of the balancer's welded beam wall at its connection to the balancer's head can be reduced to 30 cm. Thus, a welded balancer will look close to an equal-strength structure in the following form (Fig.2) with the dimensions selected according to the rocking machine SKD8-3-4000, which is the most common type size and is produced in accordance with OST 26-16-08-87.

Let's check the stress in the middle of the front arm of the welded balancer. The wall height here is h=45 cm, a=30 cm, b=1,8 cm, d=1,4 cm, $\lambda = 4 \cdot 10^{-2}, \lambda_1 = 4{,}7 \cdot 10^{-2}$
Then the moment of resistance in the middle of an equal-strength welded structure of the balancer will be:

$$W_{wel}' = \frac{30 \cdot 45^2}{6}\left[(1+2\cdot 4\cdot 10^{-2})^2 - \frac{1-4{,}7\cdot 10^{-2}}{1+2\cdot 4\cdot 10^{-2}}\right] = 3030cm^3$$

Comparing with the moment of resistance in the section above the balancer support, we obtain:

$$\frac{W_{wel}}{W_{wel}'} = \frac{4140}{3030} = 1{,}38$$

I.e., the moment of resistance in the middle of the front arm of the welded balancer is 1.38 times lower, and the bending moment will be 2 times lower.
It can be concluded that there is a reserve for reduction of metal intensity due to transition to a single-beam welded design of the balancer's body.

Conclusions: At transition to a single-beam design of a serial rocking machine rocker as a result of the made calculations for comparison of moments of resistance in the middle of an equal-strength welded balancer design and in a section over a balancer support, it is received that the moment of resistance in the middle of a front arm of a welded balancer is 1,38 times lower, and the bending moment is reduced in 2 times, that allows to draw a conclusion that there is a reserve for reduction of metal intensity due to transition to a single-beam welded design of a balancer body.

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О РЕЗЕРВАХ СНИЖЕНИЯ МЕТАЛЛОЕМКОСТИ СЕРИЙНО-ВЫПУСКАЕМЫХ СКВАЖИННЫХ ШТАНГОВЫХ НАСОСНЫХ УСТАНОВОК

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АННОТАЦИЯ

Современная скважинная насосная установка (СШНУ) состоит из скважинного оборудования, включающего в себя скважинный штанговый насос (СШН), колонну насосно-компрессорных труб (НКТ), колонну насосных штанг, и наземного оборудования, включающего в себя привод скважинного штангового насоса и устьевое оборудование. Каждый из этих элементов серийно выпускается нефтяной промышленностью в соответствии с государственными стандартами. В данной статье с целью выявления возможности снижения металлоемкости серийно выпускаемых станков-качалок исследуется возможность переход к однобалочной сварной конструкции балансира станка-качалки.

Ключевые слова: станок-качалка, балансир, однобалочная конструкция, металлоемкость.

SERİYA-BURAXILAN QUYU NASOS QURĞULARININ METAL TUTUMUNUN AZALDILMASI EHTİYATLARI HAQQINDA

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XÜLASƏ

Müasir quyu nasos qurğusu quyu ağzı ştanq nasosunu, nasos-kompressor borularının sütununu, nasos ştanq sütununu və quyu ağzı avadanlığını özündə birləşdirən yerüstü avadanlıqdan ibarətdir. Bu elementlərin hər biri neft sənayesi tərəfindən dövlət standartlarına uyğun olaraq seriyalı şəkildə istehsal olunur. Bu məqalədə kütləvi istehsal olunan nasos aqreqatlarının metal sərfiyyatının azaldılması imkanlarını müəyyən etmək üçün nasos qurğusunun mancanaq dəsgahının tarazının birtirli qaynaq olunmuş konstruksiyası keçmək imkanları araşdırılır.

Açar sözlər: mancanaq dəsgahı, tarazı, birtirli konstruksiyası, metal tutumu.

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OPTİMİZATİON AND DYNAMİCS OF DİSTİLLATİON COLUMN

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ABSTRACT

Ever increasing global energy demand necessitates the need to enhance energy generation and efficient utilization. As energy prices continue to rise, energy conservation is the prime concern for every industry. Distillation is probably the most studied unit operation in terms of optimization and control. Since distillation columns are major energy-intensive units in chemical plants and given the recent increase in fuel cost, the plant engineers are obliged to find better operating strategies and control systems. One of the significant sections that can be worked upon is finding the optimum feed tray locations of distillation columns. The energy consumptions during distillation can have a strong impact on overall profitability. Hence, the optimization of a column is aimed towards greater production and an increase in quality. Since a column is multivariable system, disturbance in any parameter can affect the overall performance as well as production loss. This paper shows how an acetone manufacturing process can be conveniently simulated first to get temperature, pressure and composition profile followed by application of optimization techniques to enhance performance. It was performed by the means of using a simulator Aspen Plus®.
Keywords: distillation Simulation, column optimization, dynamic modeling, energy consumptions, energy-intensive units.

Introduction: Process industries, in pursuit of achieving their business objectives, given the continuously changing market dynamics and strong environment regulations in place, are designing plants for an optimum performance along with reduced risk of rework by employing simulation studies [1]. Simulation & control studies are widely in practice for batch and continuous chemical process operations for last two decades [2, 3]. In the last two decades there has been a remarkable increase in application of computational techniques which are readily available for engineers [4]. Owing to these advantages engineers can easily use more profound techniques of analysis and synthesis. There is a room of improvement in mathematical modeling but the dynamics of most systems can still be described effectively for engineering purpose [5]. Distillation column has been the subject of discussion for the last two decades in both the research and industrial arena for its modeling and simulation [6-12]. Optimization techniques have been applied to problems of industrial importance ever since the late 1940s. Distillation processes are most essential unit operations in chemical engineering. They are of significant importance as separation methods in chemical and petroleum industries. Chemical and petroleum industries hold a significant share in the overall world economy. Distillation processes have huge maintenance and running cost that can be greater than the overall cost of many other processes. Therefore, there should be an effective a reliable control system for efficient and safe operation of a distillation column. It presents

various challenging control problems. Distillation columns are highly multivariable and shows non-linear behaviour. Therefore, their control is not a trivial task [13]. Distillation design analysis comprises of detailed study of vapor-liquid equilibrium calculation in some case vapor-liquidliquid (phase) equilibrium (VLLE) along with tray to tray component balances when discussing about digital computations [14-16]. Dynamic simulations of distillation columns are extensively in practice to develop effective control strategies. In the present study, steady-state simulations are being performed using Aspen Plus® followed by Aspen Dynamics® simulation, licensed software of Aspen Tech®. Aspen Dynamics® is a dynamic simulator commonly used to investigate the dynamics and control of continuous processes around some steady-state design operating point. Aspen Plus enhances the process of building and running the process simulation by adding a comprehensive system of online process modelling [2,5,15,17,18]. Aspen Plus and Aspen Dynamics developed a very sophisticated simulation that involves multiple complex steps and scripts in a very lengthy design and analysis procedure. Researchers have combined the steady-state simulator Aspen Plus and Dynamics simulator Aspen Dynamics for the simulation of control strategies [13,19, 20]. One can easily predict the behavior of any process by using fundamental and engineering relationship such as mass and energy balance, phase and chemical equilibrium. Process simulation has the ability to run different cases like "what if" analysis, sensitivity analysis and to do optimization runs. Simulation can design optimized plants as well as can also increase the profitability of existing plant by doing number of runs.

Nomenclature:

V	Vapor boilup
R	Reflux flow
D	Distillate flow rate
L	Liquid flow rate
B	Bottom flow rate
α	Relative volatility
y	Vapor composition
x	Liquid composition
M	Holdups (liquid/Vapor)
h	Enthalpy (energy/mole)
NF	Feed stage

Steady-State Continuous Process Modeling Setup: Acetone is used as solvent for biological sensitive process as well as in pharmaceutical formulation. The fully instrumented control Distillation Column is a distillation column situated in Block III University Technology PETRONAS that can be selfsufficiently run using mixture of Isopropanol (IPA) and Acetone. The First Step is to set up a simulation in Aspen Plus® that has the required pieces of equipment to size the column and auxiliary equipment of desired capacity. It should be noted that in Aspen Plus flow sheet must be a continuous process with the feed and product stream. Systematic and realistic modeling has been done to develop an understanding about distillation column parameters variation to maintain product quality. The fictitious and large feed consisting of binary mixture IPA and Acetone at a known composition (mixture 30%

acetone composition) is pumped from the feed tank via Heat Recovery Exchanger into the distillation column at a desired temperature and at a desired feed point as mentioned in figure 1. The following Unit Operation Block data has been used to simulate column as mentioned in Table 1. Pilot scale distillation column block III UTP, is used to make a case study in simulation. Column comprises of 15- bubble cap tray with the size of DN 150x5.5 m (H) with approx. 35cm tray spacing. Weir height is assumed to be0.0025 m and tray pressure drop is 0.01 bar per tray. The basic Aspen Plus® RADFRAC has been selected that incorporates implicitly a condenser and a reboiler. RADFRAC calculates the T, P, x, y, V and L on every stage. It estimates the temperature profile based on bubble and dew point temperatures of feed. RADFRAC columns allows to be operated in rating or design mode. The feed conditions and other parameters are mentioned in Table 2 below. The top stage is stage 2 and the bottom one is stage 14 with the condenser and reboiler as stage 1 and 15 respectively. Total condenser has been selected to get liquid from condenser.

Thermodynamic Property Package: The choice of appropriate thermodynamic model and the accuracy of parameters are crucial for the reliability of design. The Nonrandom Two-Liquid (NRTL) Model equation of state physical has been selected as property package (base method) in Aspen HYSYS steady-state Simulation. NRTL model is generally suggested for highly non-ideal chemical systems and can be used for both VLE and LLE applications [18, 21].

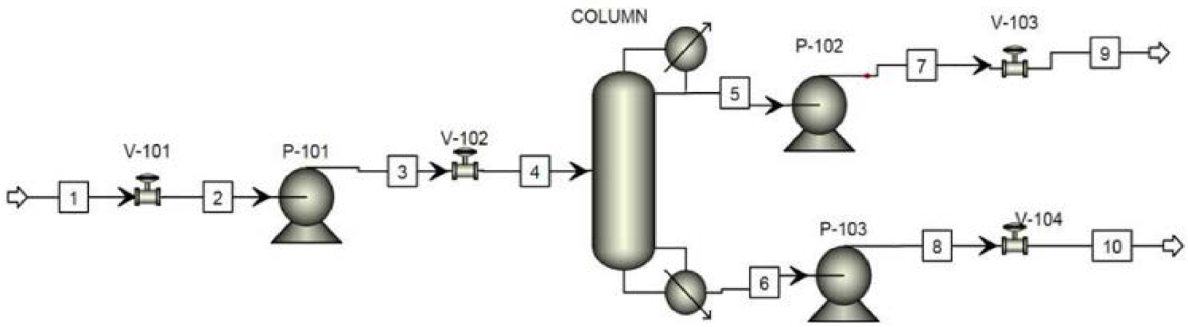


Figure 1. Aspen Plus® Steady-State Simulation

Table 1. Column/Reboiler Specification (Reference to UTP Block III distillation Column)

Distillation Column (T-100)			
Specifications	Details	Specifications	Details
Column Height	150 mm	Type of Tray	Bubble Cap Tray
Colum Diameter	5.5 m	Tray Spacing	Approx. 35 cm
No. of Trays	15-plates	Material of Construction	Stainless Steel
Feed Stage	variable	Other specifications	2 tray per section

Table 2. Feed Condition and other Specifications (Reference to UTP Block III distillation Column)

Specifications	Feed	Specifications	Feed
Temperature [C]	30.0	Reflux Ratio [molar]	3.0
Pressure [kPa]	105.6	Distillate Rate [kmole/hr]	1080
Molar Flow [kmole/h]	3600.0	Condenser Pressure [kpa]	101.3
Acetone mole fraction	0.3	Reboiler Pressure [kpa]	105.7
2-Propanol mole fraction	0.7		

Mathematical Model: A binary system (two components) with constant relative volatility throughout the column and theoretical (100 percent
efficient) trays, i.e., the vapor leaving the tray is in equilibrium with the liquid on the tray has been assumed. This means the simple vapor-liquid equilibrium relationship can be used [5];

$$y_n = \frac{\alpha x_n}{(1+(\alpha-1)x_n}$$

A single feed stream is fed as saturated liquid at its bubble point onto the tray N. Feed flow rate is F(mole/min) and composition is Z(mole fraction of more volatile component). The overhead vapor are totally condensed in a condenser and flows into reflux drum, whose holdup is MD (moles). The contents of the drum are assumed to be perfectly mixed with composition *x*D. The liquid in the drum is at its bubble point. Reflux is pumped back to the top tray (NT) of the column at rate R. Overhead
distillate product is removed at rate D. At the base of the column, liquid bottoms product is removed at a rate B and with a composition *x*B. Vapor boilup is generated in a thermosiphon reboiler at a rate V We will assume that the liquids in the reboiler and in the base of the column are perfectly mixed together and have the same composition *x*B and total holdup MB (moles).

Condenser and Reflux Drum: Component continuity (more volatile component):

$$\frac{d(M_D x_D)}{dt} = V y_N - (R + D) x_D$$

Reboiler and Column Base:

Total Continuity:

$$\frac{dM_B}{dt} = L_1 - V - D$$

Component Continuity:

$$\frac{d(M_B x_B)}{dt} = L_1 x_1 - V y_B - B x_B$$

The steady state operation of each module comprises of following equations, commonly referred to as the MESH equations. [MESH = material balance equations, efficiency relations, summation equation, and heat (enthalpy) balance equations].

Material balance equation:

$$L_{i+1} + V_{i-1} - L_i - V_i = 0$$

Stage efficiency relations:

$$y_i - y_{i-1} = \eta_{ij}[y_i^*(x_i, T_i, p_i) - y_{i-1}]$$

Where, $y_i = \frac{v_i}{V_i}$ and $x_i = \frac{l_i}{L_i}$

Summation equation:

$$L_i = \sum_{j=1}^{NC} l_{ij}$$

$$V_i = \sum_{j=1}^{NC} v_{ij}$$

Enthalpy balance equation:

$$L_{i+1}h_{i+1} + V_{i-1}h_{i-1} - L_i h_i - V_i h_i = 0$$

Assumptions: The distillation column consists of non-ideal column with following assumptions;

- The Liquid on the tray is perfectly mixed and incompressible.
- Tray vapor holdups are negligible.
- Dynamics of the condenser and the reboiler is neglected.
- Vapor and liquid are in thermal equilibrium (same temperatures) but not in phase equilibrium.
- The column is assumed to be completely insulated.

Column Optimization Parameters: Energy reduction in chemical processes is essential for energy optimization and cost effective production. However, distillation is highly energy intensive and can consume nine-tenths of the total energy in a typical chemical or petrochemical process. In most distillation column the major operating cost is reboiler energy consumption. In simulation a design specification has been used to maintain the top and bottom composition of column to achieve maximum purity. This can be done by using optimization approach for column feed location, minim no. of trays and minimum reflux ratio. Selecting optimum feed location is critical to maximizing distillation column performance. Improper feed location of a distillation column can downgrade column performance; the degree of separation is decreased at the same reflux/boil up ratio or the higher reflux/boil up ratio is required to maintain the degree of separation [21]. The objective is to minimize the reboiler heat duty which accounts for a major portion of operating costs. The optimization problem is therefore given by,

Objective function 1:

$$Minimize\ Q_B$$

Subject to:

$$8 \le N_F \le 12$$
$$L_N, V_N, T_N \ge 0$$
$$Q_B, Q_C, D, B \ge 0$$
$$N = 15$$
$$x_F = 0.3$$
$$x_D = 0.98, \quad y_D = 0$$
$$\Delta P_i = 0.01\ bar,\ 0 < n \le 15\ \text{(Integer)}$$
$$\text{L/D} = 3.0$$
$$\text{D} = 1080 \frac{\text{kmole}}{\text{hr}}$$
$$\text{P}_1 = 101.3\ \text{kpa}, \quad \text{P}_{15} = 105.7\ \text{kpa}$$

Objective function 2:

$$Minimize\ R$$

Subject to:

$$15 \le N \le 45$$
$$L_N, V_N, T_N \ge 0,$$
$$Q_B, Q_C, D, B \ge 0$$
$$N_F = 2N/3$$
$$x_F = 0.3$$
$$x_D = 0.98, \quad y_D = 0$$
$$\Delta P_i = 0.01\ bar,\ 0 < n \le 15\ \text{(Integer)}$$
$$\text{L/D} = 3.0$$
$$\text{D} = 1080 \frac{\text{kmole}}{\text{hr}}$$
$$\text{P}_1 = 101.3\ \text{kpa}, \quad \text{P}_{15} = 105.7\ \text{kpa}$$

Aspen Dynamics: For the development of control strategy, it requires the conversion of steady state model in a dynamic one. The model developed in Aspen Plus is exported to Pressure-Driven simulation in Aspen Dynamics. Before exporting to Aspen dynamics, the size and the initial specification each equipment must be identified that includes (valve pressure drops, pumps suction or discharge pressure etc. In Aspen Plus column dynamics the reflux drum is size to have a diameter of 4.08 m and length is 8.16 m
and the sump is sized to have a diameter of 5.08 m and height is 10.16 m. In column hydraulics, column diameter, tray spacing and weir height have been mentioned to complete the geometry of distillation column. It should be noted that pumps are not used to on the streams whose flow rates will be set to zero because error message will appear in Aspen Dynamics when the flow through a pump is zero. At this stage the file is now ready to be checked as a realistic pressure-driven dynamics simulation can be studied. This can be done by clicking pressure checker that either this file ready to move to Aspen Dynamics or not. In case of correctly defined plumbing, the file is ready to export into Aspen Dynamics to generate “dynf” extension file.

Initial Setup of Aspen Dynamics and Control Structure: After successfully export of file “dynf” and when it is opened in Aspen Dynamics, the first thing is to check is either integrator is running okay. There is common error message which is in practice can be noted as “Failure to initialization”. This is because of plumbing error, so there must be some correction in Aspen Plus® Steady State to configure the control structures appropriately. A larger diameter column with 15 trays would take some time to send vapor at the top of the column. Once the vapor gets to the top of the upper column, it is condensed and liquid begin to fill the reflux drum. When the sufficient liquid is built up to satisfy pump NPSH requirement, then the liquid in the reflux drum is maintained at low level. The initial flow sheet that opens up in Aspen Dynamics is shown in Fig. 2. A default controller has been automatically installed with

the closed loop process flow diagram i.e. pressure controller at the condenser, in control configuration panel it can be seen that the default for the pressure ranges are from (0-2 bar) and this controller is maintaining the controller pressure by manipulating the condenser heat removal (direct Qc) The Aspen Plus default tuning parameter are used (Kc =20 and τ_I= 12 min).

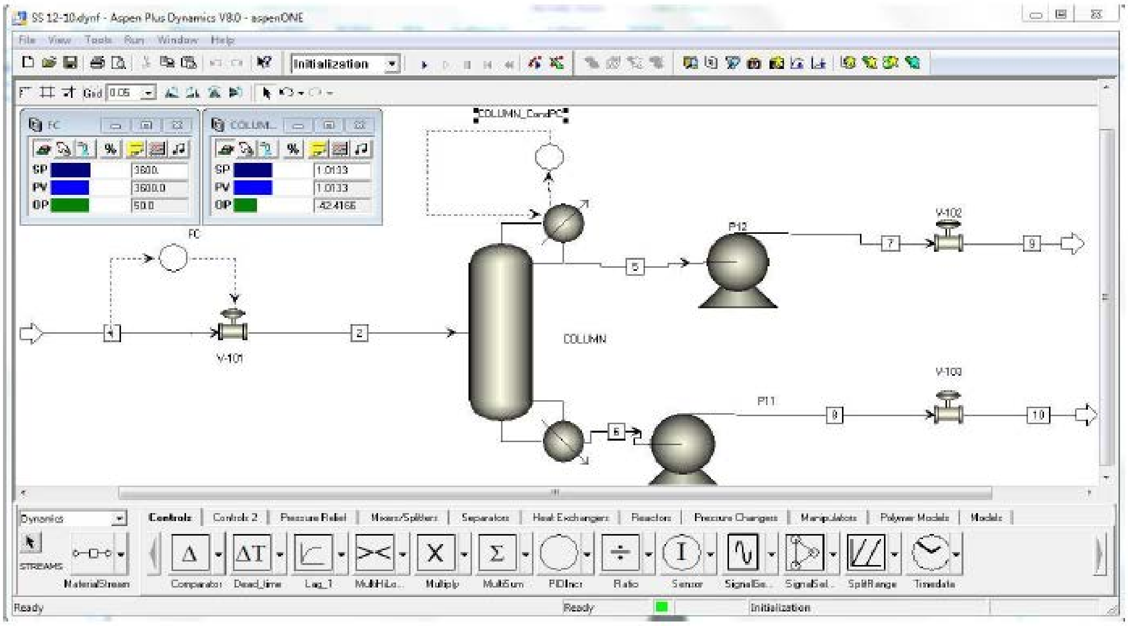


Figure 2. Initial Aspen Dynamics flowsheet after exporting.

Results and Discussions: Steady State Simulation. First Step of this study was the analysis of a given binary mixture system, with a defined physical property package. Aspen physical property system has a large no. of built-in binary parameter for different models. The column specification diagram i.e. temperature, pressure, composition can easily be generated in Aspen with the calculation of phase equilibrium relationship that are all hidden is Aspen data bank. The column temperature and composition profile can be obtained by selecting block column and plot the profile as mentioned in Figure 3. The composition of Acetone can be noted as 97% pure at top of the column at reflux ratio 3 molar.

Optimum Feed Tray and Minimum Conditions: Once the pressure, temperature and composition profile has been determined along with mole purity has been set to the desired value, so we can find the optimum feed tray. In addition, the minimum reflux ratio and the minimum no. of stages can also be determined to run the column at optimal conditions. The Simulation is run at fixed top and bottom composition to get the optimum feed tray.
As shown in Fig. 4, It can be seen that feed on stage 10 gives the minimum energy consumption i.e. 40.8 MW to achieve 98% Acetone as distillate. The simulation can be used to find the minimum reflux ratio by increasing the no. of stages until there is no further reduction in reflux ratio as shown in fig. 5. As product purities are kept constant. It is assumed that the feed stage is a fixed ratio of total no. of stages.

Conclusion: This article presents the optimization and dynamics of distillation column for acetone production unit. The design flow sheet is simulated using NRTL thermodynamic model which describes vapor liquid equilibrium data appropriately. It can be witnessed that Aspen Plus and Aspen Dynamics is a comprehensive tool which can be used to optimize the column and can be used to develop different control strategies. An appropriate control structure can be modeled from Aspen Dynamics control model library to develop a realistic scheme that will be helpful to understand physical reality.

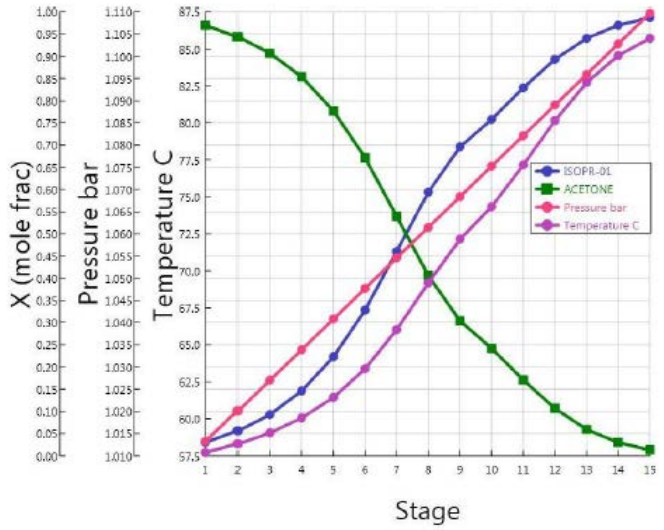


Figure 3. Temperature, pressure and composition profile of column.

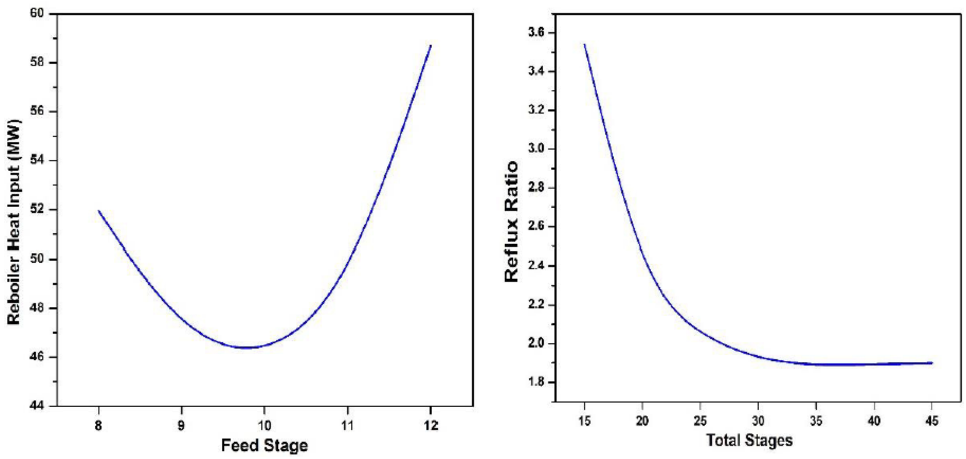


Figure 4. Effect of Feed Stage on Reboiler Heat Input , effect of Total Stages on . Minimum Reflux ratio

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ОПТИМИЗАЦИЯ И ДИНАМИКА РЕКТИФИКАЦИОИНОЙ КОЛОННЫ

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РЕЗЮМЕ

Растущий мировой спрос на энергию требует увеличения её производства и эффективности использования. Поскольку цены на энергоносители продолжают расти, энергосбережение становится ключевым вопросом для любой отрасли. Перегонка, пожалуй, наиболее изученная операция с точки зрения оптимизации и контроля. Поскольку ректификационные колонны являются основными энергоемкими устройствами на химических предприятиях, а также с учетом недавнего роста цен на топливо, инженеры предприятий вынуждены искать лучшие стратегии эксплуатации и системы управления. Одним из важных участков, над которым можно поработать, является определение оптимального расположения тарелок ректификационных колонн. Энергозатраты при дистилляции могут иметь сильное влияние на общий прирост. Таким образом, оптимизация колонны больше направлена на повышение производительности и качества. Поскольку колонна является многофункциональной системой, сбой любого параметра может повлиять как на её, так и на общую производительность. В этой статье показано, как можно легко смоделировать процесс производства ацетона для получения различных профилей температуры, давления и состава с последующим применением методов оптимизации для повышения производительности. Это было сделано с помощью программного обеспечения Aspen Plus®.

Ключевые слова: моделирование дистилляции, оптимизация колонны, динамическое моделирование, энергопотребление, энергоемкие агрегаты.

REKTİFİKASİYA KALONUNUN OPTİMİZASİYASI VƏ DİNAMİKASI

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XÜLASƏ

Enerjiyə artan qlobal tələbat enerji istehsalının artırılması və enerjidən səmərəli istifadəni zəruri edir. Enerji qiymətləri artmağa davam etdikcə, enerjiyə qənaət hər bir sənaye üçün əsas məsələdir. Distillə, yəqin ki, optimallaşdırma və nəzarət baxımından ən çox öyrənilən əməliyyatdır. Rektifikasiya kalonları kimya zavodlarında əsas enerji tələb edən qurğular olduğundan və yanacaq qiymətlərinin son artımını nəzərə alaraq, zavod mühəndisləri daha yaxşı əməliyyat strategiyaları və idarəetmə sistemləri tapmağa məcburdurlar. Üzərində işlənə biləcək əhəmiyyətli bölmələrdən biri rektifikasiya kalonlarının optimal boşqab yerlərinin tapılmasıdır. Distillə zamanı enerji istehlakı ümumi qazanca güclü təsir göstərə bilər. Beləliklə, kalonun optimallaşdırılması daha çox istehsala və keyfiyyətin artmasına yönəldilmişdir. Kalon çoxfunksiyalı sistem olduğundan, hər hansısa parametrin pozulması istehsal itkisi ilə yanaşı ümumi performansa da təsir edə bilər. Bu məqalə, temperatur, təzyiq və kompozisiya profilini əldə etmək üçün əvvəlcə aseton istehsalı prosesinin rahat şəkildə necə simulyasiya oluna biləcəyini, daha sonra performansı artırmaq üçün optimallaşdırma üsullarının tətbiqini göstərir. Bu Aspen Plus® proqramından istifadə etməklə həyata keçirilmişdir.

Açar sözlər: distillə simulyasiyası, kalonların optimizasiyası, dinamik modelləşdirmə, enerji istehlakı, intensiv enerji aqreqatları.

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INVESTIGATION OF MAGNETIC EQUIPMENT IN BALL VALVE

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ABSTRACT

The article is aimed at solving the problems of sealing of ball valves with the use of metal caps with a magnetic device. It was determined that the modern level of structural design of ball valves is reaching its perfect level. However, in order to bring the valves to a modern level in terms of material supply, it is important to develop metal seals made of lightweight materials that are resistant to corrosion at low or high temperatures.
Keywords: ball valve, sealer, magnetic device, sealing, metal sealing.

Introduction: Valves are used in any field of industry. They are made according to the requirements of the fields in which they are used. Valves are often exposed to high temperatures, pressures and corrosive substances. As a result, they sometimes fail to work for days or even hours. Therefore, research is constantly being carried out to improve these structures, and innovations are being sought. The main purpose of this article is to increase the sealing ability of the ball valve. A special magnetic device was used for this purpose. [1]
The sealing element is a part installed on the crane. When the tap is closed, the clamp seals the crane. Sealants are usually made of plastic or rubber, and they break down quickly when exposed to high pressure and temperature. The purpose of this article is to design new metal caps for ball cranes. [2]
Valves are devices that control the flow of fluid. Valves developed with the help of new technologies can control not only the flow, but also the pressure, speed and direction of flow. When the valve is closed, it cuts off the flow, otherwise it allows the flow to pass. . The element that creates tightness in faucets is the valves. Rubber, plastic, teflon and nylon materials can be used to ensure reliable sealing. The advantage of these materials is their high elasticity. However, these materials can be deformed under high pressure, temperature or high corrosive environments. This makes it difficult to seal effectively. The use of metal caps has a special place in overcoming this problem. The main fundamental requirements are the proper operation of the crane in high temperatures, pressures and corrosive environments.
The main issues to consider when designing a valve:
1) Quality
2) Can be produced
3) Must be economically profitable
4) It should be easy to install and repair.
The use of metal for sealing is one of the best ways to obtain the items listed above. Applies to mechanical design technology:
1) Application of innovations
2) Planning

3) Production
4) Testing process

The purpose of the study: When using a magnetic device and metal seals on a ball valve, to determine the optimal geometric shape of the magnetic device for maximum sealing.

Ways to solve the problem: Valves are devices that operate under pressure and cut off the flow, allow the passage, as well as change the direction of flow. All this is done through the connecting element of the flow (sphere, gate, etc.). There are two main types of cranes: linear and rotary. Spherical cranes are one of the rotating motion cranes. The spherical valve is shown in Figure 1. The sphere rotates between the caps. The main advantages of ball valves are that they do not require lubrication and do not require high torque to operate. The sealer are part of the faucet body and ensure the tightness of the mouth that controls the flow. The sealing ring has special material properties and provides a contact surface for the fastening element. Soft materials such as plastic, Teflon and nylon are used as sealing materials due to their elasticity. One of the most important technical indicators of sealants is their service life. However, these materials can be deformed under high pressure, temperature or high corrosive media. This makes it difficult to seal effectively. The use of metal sealing elements has a special place in overcoming this problem. The manufacture and installation of metal caps must be carried out accurately to achieve good reliability. However, the application of metal sealants requires high funding. Also, in highly corrosive environments, metal sealants can fail quickly. Along with this problem, the fact that metal sealants increase the total mass should not be overlooked. [5]

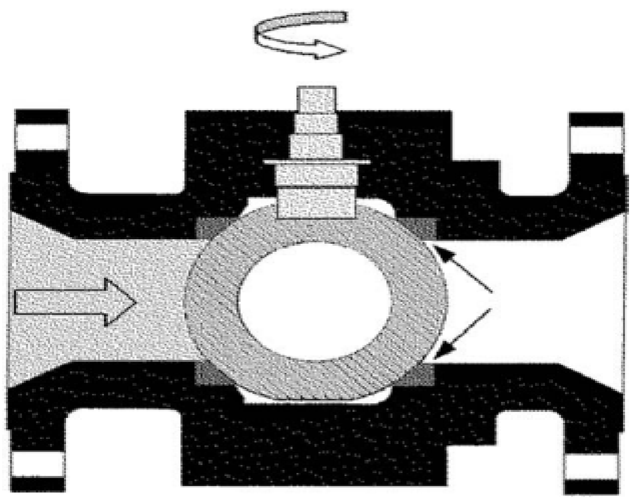
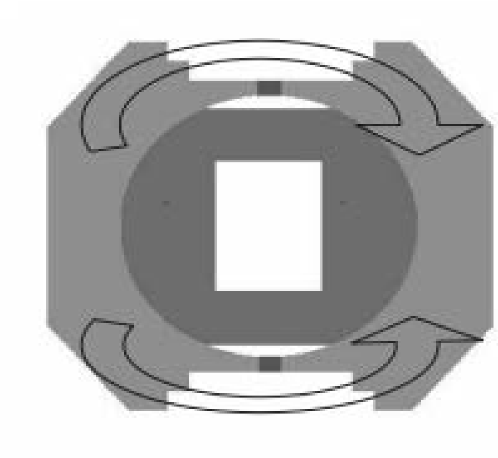


Figure 1. Ball valve cross section view.

When using metal sealants, high sealing occurs between metal-to-metal surfaces. In this case, the torque required to rotate the sphere is higher. It is used in three activators to overcome this problem. Designing metal seals to create a tight connection with the fastener is not an easy

process. Therefore, the main purpose of this scientific work is to develop high-cost metal sealants that work in high temperatures and corrosive environments, as well as can be easily installed and repaired. The valve must also be operated at low torque and must be able to open and stop the flow immediately. When designers have a concept, the goal is to build the final design structure according to technical and economic criteria. The initial design leads to the refinement of the plan. The arrangement, types, sizes and materials of all the individual parts are recorded at the final design stage. The seals on the ball valves are always in contact with the ball. [6]

Figure 2. Open position of ball valve.



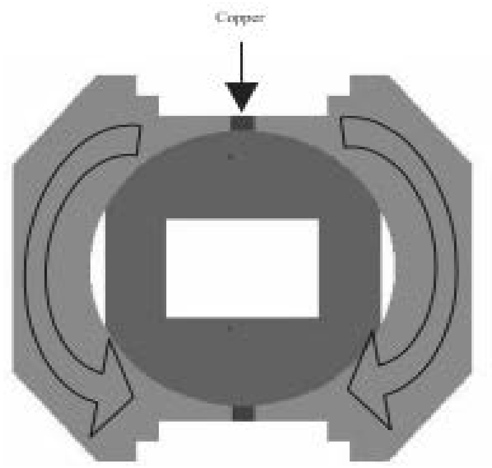


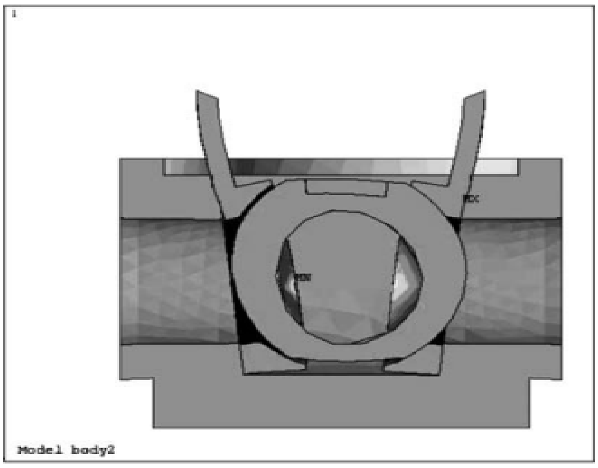
Figure 3. Close position of ball valve.

When using magnetic parts for sealing, the magnetic particle uses the magnetic force for sealing when the tap is closed. It is not affected by magnetic forces when the valve is closed.

If the magnetic field acting on the valve when it is open does not affect it when the valve is closed, then the frictional force acting on the valve is small. Figures 2 and 3 show a magnetic device.

The central part of the magnetic device is made of magnet, and the other outer part is made of steel and copper. Steel is one of the pure magnetic conductive materials. Copper does not have magnetic permeability. Magnetic waves pass through steel more easily than copper. [7]
Once the design of the magnetic device is complete, the next task is to connect the device to the seat. The seat is a circular ring and it should fit the back. To absorb the magnetic force, a tongue-shaped part extending from the edge of the seat is added
The seat can now receive magnetic force in the closed position, but not in the open position. To ensure sealing, the seat must be in close contact with the ball and the lid body in the closed position. Therefore, the part of the body holding the seats and the passage of the sphere are V-shaped. Therefore, the horizontal magnetic force applied by the magnetic device pushes the seats down due to this inclination, squeezing the ball and the seats together to ensure sealing. [8]

Figure 4. Material shape changing after high pressure fluid affect.



From Figure 5 we can see that the contact elements are still in strong contact with each other and the gaps between the contact elements are zero. This means that the ball valve is still sealed under pressure. The magnetic force applied to the seats after the magnet is set depends on the shape of the magnetic device. Therefore, the purpose of optimization here is to increase the magnetic force by adjusting the position and size of the magnetic device. Because of the limited area of the cover and the inability to change the height of the magnetic device, the mouthpiece does not affect the magnetic lines. The model can be changed in two dimensions to optimize the design. The limitation in the design is the cover area, the magnetic device must be smaller than the cover area. The goal of optimization here is to obtain the highest magnetic intensity-H, as this maximizes the magnetic force.

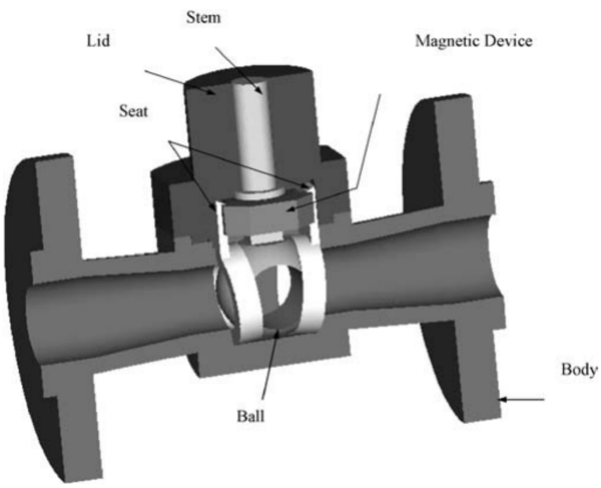
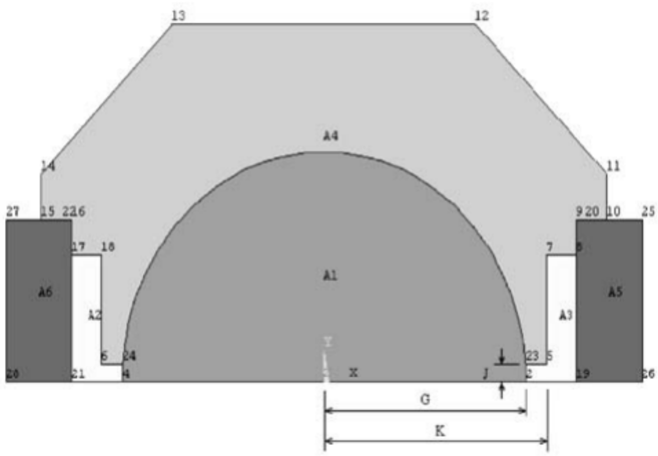


Figure 5. Sealing materials under magnetic affect.

Figure 6. Geometry of magnetic device.



The ANSYS optimization tool was used for optimization. This tool allows us to choose the best settings for the design.

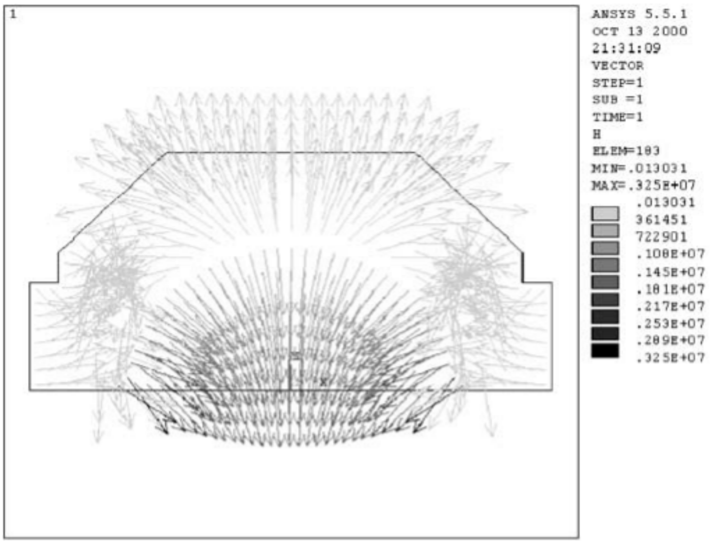


Figure 7. Before optimization.

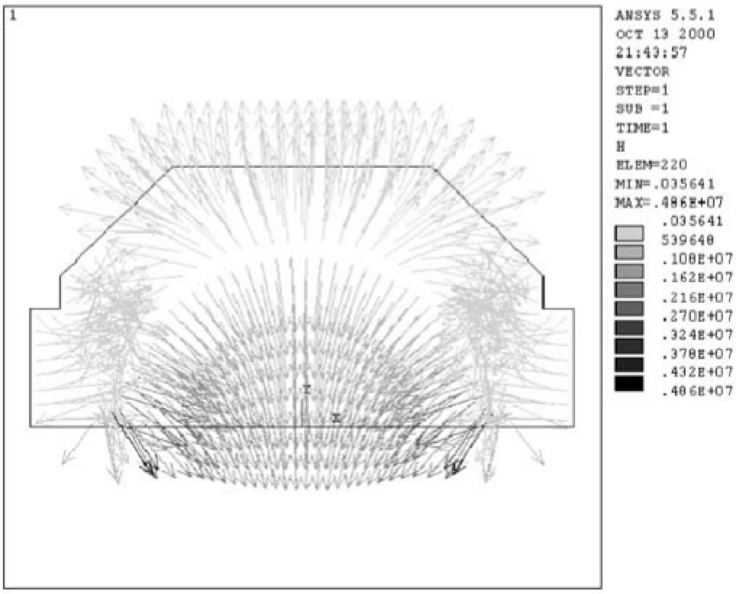


Figure 7. After optimization.

Different size of magnetic device
The evaluation of the optimization of the design of spherical cranes is given in Table 1.

Table 1. Optimization evaluation

	J	K	G	H
First model	1.5	20	22	$0.43886x10^{7}$
Optimal version	2.102	21.5	22.476	$0.62325x10^{7}$

Processing of results: We can see that the contact parts of the ball valve are still in strong contact with each other during the process and the gaps between the contact elements are zero. This means that the ball valve is still sealed under pressure. The
magnetic force applied to the seats after the magnet is set depends on the shape of the magnetic device. Therefore, here we can see that the magnetic force increases by adjusting the position and size of the magnetic device after optimization. As it is not possible to limit the area of the cover and change the height of the magnetic device, it is known that the mouthpiece does not affect the magnetic lines. The model is modified in two dimensions to optimize the design. The limitation in the design is the cover area, the magnetic device must be smaller than the cover area.

Conclusions: The use of a magnetic device with metal caps allows ball valves to operate in high temperature and pressure conditions
Under the influence of the magnetic force created by the application of a magnetic device, the sealing capacity of metal seals will be increased by 1.4 times.

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MAQNİT CİHAZLI KÜRƏVİ KRANLARIN TƏDQİQİ

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XÜLASƏ

Məqalə kürəvi kranlarda maqnit cihazılı metal kipgəclərin istifadəsilə onlarda yaranan kipləşdirmə problemlərinin həllinə yönəlmişdir. Müəyyən edilmişdir ki, kürəvi kranların konstruktiv quruluşunun müasir səviyyəsi özünün mükəmməl səviyyəsinə çatmaqdadır. Buna baxmayaraq kranların müsir səviyyəsinin yüksəldilməsi üçün onlarda aşağı və ya yüksək temperturlarda yeyilməyə davamlı, yüngül çəkiyə malik materiallardan düzəldilmiş metal kipləşdiricilərin hazırlanmasına böyük ehtiyac vardır. Məqalədə yeni maqnit cihazlı kürəvi karanların işlənməsinin elmi yanaşması verilmişdir.Məlum olmuşdur təqdim olunmuş kürəvi kranlarda maqnit cihazlı metal kipgəclərin istifadəsi kürəvi kranların yüksək temperatur və təzyiq şəraitində işləməsinə şərait yaradır və metal kipləşdiricilərin kipləşdirmə qabiliyyəti artır.

Açar sözlər: kürəvi kran ,kipləşdirici , maqnit cihazı, kiplik, metal kipləşdirici.

ИССЛЕДОВАНИЕ МАГНИТНОГО ОБОРУДОВАНИЯ В ШАРОВОМ КРАНЕ

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АННОТАЦИЯ

Статья направлена на решение проблем герметизации шаровых кранов с применением металлических колпачков с магнитным устройством**.** Определено, что современный уровень конструктивного исполнения шаровых кранов достигает своего совершенного уровня. Однако для вывода арматуры на современный уровень по материалообеспеченности важно разработать металлические уплотнения из легких материалов, устойчивых к коррозии при низких или высоких температурах.

Ключевые слова: кран шаровой, уплотнитель, магнитное устройство, уплотнение, металлическое уплотнение.

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A BRIEF OVERVIEW OF VACUUM WATER RING PUMPS AND A THEORETICAL MODEL OF THE PERFORMANCE OF THESE PUMPS

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ABSTRACT

Practice shows that vacuum water ring pumps are widely used in industry. Such pumps are used in almost all areas of the oil industry, metallurgy, chemical industry, mechanical engineering, food industry, agriculture, pulp and paper industry, gas and many other areas. Taking into account all the above-mentioned areas of application, the question arises about increasing the service life and reliability of LCVs (liquid water ring pumps). This article provides a fundamental study of the theory of operation and performance of a liquid ring pump.

Keywords: vacuum pump, theoretical model, coolant, foaming, cavitation, actual performance.

Water-ring vacuum pumps are such hydraulic machines that are used for pumping gas mixtures, as well as for the formation of airless (vacuum) media. These pumps can pump out large volumes of gases. At the same time, gas mixtures are cleaned of dust and other impurities during pumping. Depending on the chemical composition of the pumped gaseous substances, KVN (liquid vacuum pumps) can be made of various materials. And when it is necessary to additionally purify gases, two-stage KVN pumps are used. The use of two-stage ZHVN leads to increased pressure when exiting the pump outlet pipe.

Within the framework of a new alternative concept, the method of using various polymer additives is considered, which becomes particularly relevant when pumping heavily polluted gases. The effect of polymers on the pump operation in the case of pumping a gas mixture is considered.

Liquid vacuum water ring pumps are made of various metal alloys. And accordingly, for each of the existing types of ZHKVN, you should use your own type of polymer additives. In this article, a theoretical research method was used. The typical design of the working area and the principle of operation of the pump are considered. Also, information was found from open sources about various polymer additives that are used in the ZHKVN for addition to the working fluid. Their main varieties and the properties they impart to the working fluid are described. Their influence on the pump body is investigated.

The principle of operation of the pump is based on the formation of a liquid ring inside the working area. Ordinary water is used as a liquid in water-ring vacuum pumps. Inside the housing there are blades located with an offset relative to the center of the housing. During the operation of the pump, the blades transfer kinetic energy to the water and form a water

wheel, the liquid, with each rotation of the impeller, displaces gas into the outlet and, exiting from the resulting vacuum, draws gas from the inlet.
Liquid water ring vacuum pumps have some advantages and disadvantages.
The advantages of such pumps include:

- the output gases pumped have no oils in their composition;
- gases are cleaned of mechanical impurities;
- easy maintenance, repairs and increased reliability;
- more affordable compared to other types of pumps with similar characteristics;

Disadvantages include:

- it is required to constantly clean the liquid from solid particles that are deposited during the operation of the pump;
- systematic replacement or recirculation of the coolant is required in order to reduce the temperature in the system;
- cavitation formation inside the pump;
- performance depends on the temperature of the antifreeze liquid and gases.

The traditional theory of a liquid ring pump was based on an ideal working cycle "suction-compression-injection" which consisted of three processes.

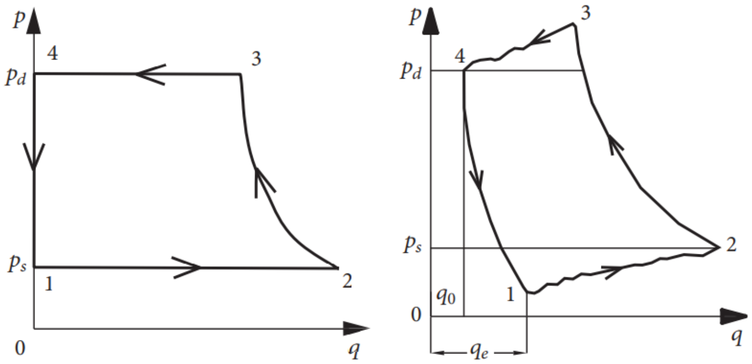


Figure 1. Diagram - of the gas working cycle in a liquid ring pump.

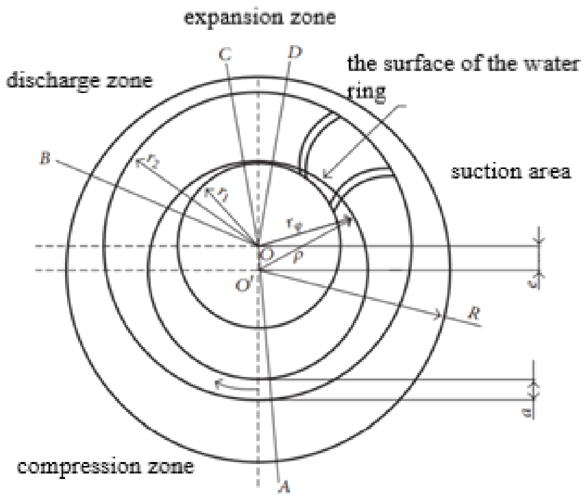


Figure 2. Geometric parameters of the liquid ring pump.

The residual gas and its expansion process were not taken into account in the theoretical model of the ideal working cycle of a liquid ring pump, which led to a discrepancy between theoretical and practical results.
Therefore, our theoretical model should complement the expansion process based on the ideal operating cycle model. For the integrity of the model, the formulas in each zone are given in detail below.
Suction area. The determination of the geometric parameters of the water ring pump is shown in Fig. 3. The distance from any point on the surface of the water ring in the suction zone to the center of the impeller is expressed as

$$r_\varphi = r_2\sqrt{\frac{2(R-p)}{\mu r_2} + v^2} \tag{1}$$

Where

$$\rho = r_2\sqrt{1 + \left(\frac{e}{r_2}\right)^2 + 2\frac{e}{r_2}\cos\varphi} \tag{2}$$

The theoretical suction power q_{th} is defined as the volumetric flow rate of gas through the OA section under the assumption that the liquid-ring pump makes full suction:

$$q_{th} = \frac{1}{2}\mu\omega b r_2^2\left[(1-\alpha)^2 - v^2\right] \tag{3}$$

Compression zone. The distance from any point on the surface of the liquid ring in the compression zone to the center of the impeller is equal to

$$r_\varphi = r_2\sqrt{\frac{(1-\alpha)^2 + (\sigma-1)v^2}{\sigma}} \tag{4}$$

The compression ratio of the pressure at the circumferential angle to the suction pressure must be solved by an algebraic equation:

$$\frac{1}{\sigma^2} = A^2\left(1 - \frac{\sigma-1}{\varepsilon}\right) \tag{5}$$

Unloading area. The distance from any point on the surface of the liquid ring in the discharge zone to the center of the impeller is expressed as

$$r_\varphi = r_2\sqrt{v^2 + \frac{2(R-\rho)}{\mu r_2}\sqrt{\frac{1}{3}\left(1+\frac{1}{\varepsilon}\right)}} \tag{6}$$

Expansion zone. The distance from any point on the surface of the liquid ring in the expansion zone to the center of the impeller is also calculated by (6). Therefore, the full shape of the surface of the liquid ring in the liquid ring pump can be obtained in accordance with (1), (4) and (6). The performance of the gas returning to the suction zone of the liquid ring pump pump, calculated by the formula

$$q_0 = \mu \int_{r_1}^{r_{\varphi=2\pi}} \omega r b\, dr = \frac{1}{2}\mu\omega b\left(r_{\varphi=2\pi}^2 - r_1^2\right) \tag{7}$$

where $r_{\varphi=2\pi}$ the distance from the surface of the liquid ring at a circumferential angle φ = 2π to the center of the impeller. The gas volume q_0 changes to q_e after the polytropic expansion process, and then enters the next duty cycle:

$$q_e = q_0\left(\frac{p_d}{p_s}\right)^{1/m} \tag{8}$$

Therefore, the actual suction capacity of the liquid ring pump is expressed as

$$q_s = \left\{\frac{1}{2}\mu\omega b r_2^{\alpha}\left[(1-\alpha)^2 - v^2\right] - q\left(\frac{p_d}{p_s}\right)^{1/m}\right. \tag{9}$$

Then the actual throughput can be calculated by the process of isothermal compression:

$$qd = \frac{q_s P_s}{p_d} \tag{10}$$

The effective capacity of a liquid ring pump is usually calculated according to the isothermal compression capacity. However, an accurate prediction of shaft power is important for evaluating overall efficiency and exploring ways to reduce energy consumption in a liquid ring pump. The shaft power mainly consists of two parts: the total gas power consumption and the friction losses of the working fluid. The total power consumption of the gas is equal to the work of adiabatic compression:

$$N_g = \frac{k}{k-1} P_s q_{th}\left[\left(\frac{p_d}{p_s}\right)^{(k-1)/k} - 1\right] \tag{11}$$

The empirical formula for liquid friction losses in a liquid-ring pump for turbulent flow was given by Praeger:

$$N_s = 0.708\frac{\rho}{2}\omega^3 r_2^5 Re^{-0.1732}$$

$$Re = \frac{\omega r_2^2}{v}$$

It is established that the axial width of the impeller is usually large in liquid-ring pumps, but has not yet been taken into account in the Praeger equation (12). Referring to the formula of the friction power of the disk in the centrifugal pump, equation (12) of the liquid friction loss can be corrected as

$$N_l = 0.708\frac{\rho}{2}\omega^3 r_2^5 Re^{-0.1732}\left(1 + f\frac{b}{r_2}\right) \tag{14}$$

Therefore, the power on the shaft of the water ring pump is written as

$$N = N_g + N_l = \frac{k}{k-1} P_s q_{th}\left[\left(\frac{p_d}{p_s}\right)^{(k-1)/k} - 1\right] + 0.708\frac{\rho}{2}\omega^3 r_2^5 Re^{-0.1732}\left(1 + f\frac{b}{r_2}\right) \tag{15}$$

Global efficiency is finally evaluated by the following expression:

$$\eta = \frac{p_s q_s \ln(p_d/p_s)}{N}$$

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HƏLQƏLİ SU NASOSLARININ QISA TƏSVİRİ VƏ BU NASOSLARIN MƏHSULDARLIĞIN NƏZƏRİ MODELİ

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XÜLASƏ

Təcrübə göstərir ki, vakuum həlqəsi nasosları sənayedə geniş tətbiq olunmuşdur. Belə nasoslar neft sənayesinin, metallurgiyanın, kimya sənayesinin, maşınqayırmanın, yeyinti sənayesinin, kənd təsərrüfatının, sellüloz-kağız sənayesinin, qaz sənayesinin və bir çox başqa sahələrdə demək olar ki, bütün sahələrində istifadə olunur. Yuxarıda göstərilən bütün tətbiq sahələrini nəzərə alaraq, MHSN (maye həlqəli su nasosları) xidmətinin və etibarlılığının artırılması məsələsi ortaya çıxır. Bu məqalədə maye- həlqəli nasosun iş nəzəriyyəsinin və məhsuldarlığının fundamental tədqiqi aparılır.
Açar sözlər: vakuum nasosu, nəzəri model, soyutma mayesi, köpüklənmə, kavitasiya, faktiki məhsuldarlıq.

КРАТКИЙ ОБЗОР ВАКУУМНЫХ ВОДОКОЛЬЦЕВЫХ НАСОСОВ И ТЕОРЕТИЧЕСКАЯ МОДЕЛЬ ПРОИЗВОДИТЕЛЬНОСТИ ДАННЫХ НАСОСОВ

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АННОТАЦИЯ

Практика показывает, что вакуумные водокольцевые насосы получили широкое применение в промышленности. Такие насосы используют практически во всех областях нефтяной промышленности, металлургии, химической промышленности, машиностроении, пищевой промышленности, сельском хозяйстве, целлюлозно-бумажной промышленности, газовой и во многих других сферах. Учитывая все вышеперечисленные сферы применения, возникает вопрос об увеличении срока службы и надежности ЖКВН (жидкостных водокольцевых насосов). В данной статье проводиться фундаментальное исследование теории работы и производительности жидкостно-кольцевого насоса.
Ключевые слова: вакуумный насос, теоретическая модель, охлаждающая жидкость, пенообразование, кавитация, фактическая производительность.

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STUDY OF THE EFFICIENCY OF HYDROMECHANICAL PACKERS USED IN THE OPERATION OF DELICATED WELLS

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ABSTRACT

The article considers the general shortcomings of packers designed for carrying out geological and technical measures in an open hole. These packer deficiencies can cause problems in the operation of wells, which leads to an increase in non-productive time or even to the loss of a well and, as a result, leads to high costs for oil and gas companies. The design of sliding support elements, centering the structure in the open hole, preventing leakage of the sealing rubber element due to the occurrence of a pressure drop, and subsequently ensuring a reliable packer break, has been studied.
Key words: hydromechanical packer, wellbore, sealing element, the packer unit.

Introduction. The performance indicators of packer systems used in an open hole to eliminate mud loss intervals include: tightness of the isolated interval under differential pressure; axial activation force of the packer, the number of repeated isolations of well intervals (up to 10 landings) in one round – trip operation. A great contribution to solving the problem of developing packer systems was made by teams of industry research institutes of the years. However, the existing problems associated with a decrease in operational efficiency with multiple repetitive packer landings remained unresolved.

Relevance: It is known from oilfield engineering that in order to ensure multiple repetitive landings in well systems, it is desirable to use multiple action mechanisms using collet elements as basic units, and to obtain tight isolation of well intervals during repeated landings, it is necessary to use mechanisms similar to sliding support assemblies. However, the effect of multiple landings on the operation of these nodes and their operational efficiency in packer systems has not been studied.
Thus, the task of developing, testing and introducing the packer systems into production for well interventions (GTO) in an open hole, allowing to achieve the necessary combination of reliability and efficiency of the basic units, is relevant and is determined by the growth of the technical level of equipment that eliminates complications, associated with the absorption of drilling fluid during drilling, development and current workover of wells.

Objective: Improving the operational efficiency of packer systems when operating in an open hole based on the improvement of basic units used in multiple-repetitive landings. To achieve this goal, the following tasks were solved:
1. Analysis of operational efficiency of existing designs of packer systems.
2. Development and study of analytical models of the stress-strain state of working elements in the basic units of the packer.

3. Experimental studies to determine the performance characteristics of the basic units of prototypes of the packer system and analysis of deviations of the obtained results from the calculated ones.
4. Analysis of the obtained patterns of changes in operational characteristics based on analytical and experimental studies of the basic units of the packer system.
5. Technical solutions based on analytical and experimental studies aimed at improving the operational efficiency of the packer system.

Over the past few years, progress in the field of hydraulic fracturing services in horizontal sections of the well has taken a sharp step forward, in addition, it should be noted that the leap occurred not only in the development of hydraulic fracturing technologies, but also a significant increase was recorded in the development of competitive equipment designs of domestic and foreign manufacturer. Such an interest in the use of MSHF technologies has generated a new branch of the development of open hole packer systems, namely hydromechanical packer systems for MSHF or annular packer devices for MSHF. Such packer systems are usually non-recoverable packer devices which are set by creating excess pressure inside the pipe string. The annular 23 multistage hydraulic fracturing packer devices are lowered into the well as part of the multistage fracturing assemblies, and one of the main criteria for such packers is their dimensions:

- outside diameter;
- internal flow channel;
- equipment length.

Each of the above criteria plays an important role in the performance of MSHF operations, therefore, the developers pursue the following goals in the design of new types of packer devices for MSHF:

- obtaining the largest flow channel to reduce hydraulic resistance and ensure the passage of a tool on coiled tubing through the packer device;
- creation of a packer device having the smallest overall dimension in length in order to ensure the maximum reduction in the distance of isolated intervals in an open hole;
- outer diameter, most often regulated by the customer.

Team Oil Tools produces a special hydraulic non-recoverable packer device for multistage hydraulic fracturing type T1019 HIP (Fig. 1.).

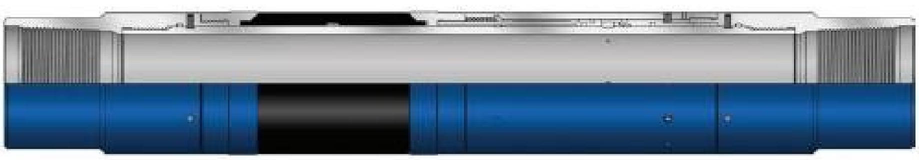


Figure 1. Packer device type T1019 HIP manufactured «Team Oil Tools» .

Type T1019 HIP Packer is an isolation hydraulic packer designed to perform reservoir isolation work in T-Frac technologies multilayer completions as part of a liner string. The packer device can run in both open hole and cased hole and can be used in vertical, directional and horizontal wells. This packer device is not used in the same system with anchor devices.

The activation unit for these packer devices is triggered by creating a pressure drop in the tubing space and does not apply to multiple action mechanisms.
The features and advantages of this type of packer devices include:

- the activation unit ensures the exclusion of unauthorized packing during the landing of the packer device;
- when the packer device is activated, only one element acts on the rubber cuffs of the piston;
- the ability to control the activation pressure of the packer (landing) before the start of the trip;
- internal diameters are identical to the diameters of the pipes, which include the packer device.

OJSC "TYAZHPRESSMASH" today is one of the leading manufacturers of equipment for multi-stage hydraulic fracturing, in addition, it has its own developed technologies for hydraulic fracturing in horizontal wells. For multistage hydraulic fracturing layouts, OJSC TYAZHPRESSMASH produces hydromechanical packer devices of the GRP-I.020 series (Fig. 2.), used to cut off hydraulic fracturing intervals.

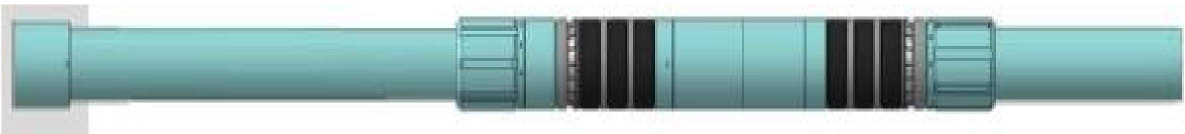


Figure 2. Scheme of the hydromechanical packer device of the GRP-I.020 series manufactured « OJSC TYAZHPRESSMASH »

Currently, interest in the development of technologies for the use of such packer systems is concentrated in the following areas:
1. Creation of designs of packer devices with a minimum outer diameter, and the largest internal flow channel.
2. Research for technologies and designs of hydraulic packer devices, with the possibility of installing them in the well due to excess hydraulic pressure for a long period of time, after which the device is torn off from the well with its subsequent extraction, that is, the introduction of wet action mechanisms into such systems.
The need to reduce the outer diameter of the packer device is due to the fact that today the MSHF liner is usually installed in the mother string with a diameter of 177 mm, and as a rule, the casing packer devices are installed in an open hole drilled with a bit with a diameter of 149 - 161 mm. The maximum outer diameter of such arrangements is typically 148 mm with an inner diameter of 100 mm. Since today oil and gas companies are striving to reduce costs and are trying to achieve this by reducing the metal consumption of the well, to accomplish this they use mother columns not of diameter 177 mm, but 168 mm. As a result, MSHF assemblies are lowered into an open hole, drilled with a bit with a diameter of 136 - 152 mm. Which leads to the need to make equipment with an outer diameter of 136 mm and an internal flow channel with a diameter of 96.7-100 mm. Many manufacturers are currently unable to

achieve the specified values and are constantly looking for new design solutions. In addition, the global oil and gas community has repeatedly voiced calls for the development of a multistage hydraulic fracturing layout with the possibility of its extraction after hydraulic fracturing. Now this method is not possible, since in the layout of the multi-stage hydraulic fracturing, as a rule, packer systems of a non-recoverable type are used, which motivates manufacturers around the world to find a constructive solution for the packer device, which can provide landing using hydraulics, maintaining the working position of the packer device after pressure relief and the possibility of failure after a certain period of time. There are currently no such packer systems in the oil industry. When designing a packer, it is necessary to take into account the possibility of its operation in deviated wells and take into account the ability of the packer to deform when passing curved parts of the wellbore. The length of the packer, at which it will touch the casing at three points (at the ends and in the middle) when it passes the curved part of the well (Fig. 3), is determined from the expression [1]:

$$l_{packer} = 2 \cdot \sqrt{2R\Delta s} \quad (1)$$

where R - is the radius of the wellbore bend;
Δs - clearance between packer and casing.

$$l_{packer} = 2 \cdot \sqrt{2R\Delta s} = 2 \cdot \sqrt{2 \cdot 50 \cdot 0{,}016} = 2{,}5 \text{ m.}$$

With a longer length, the packer, passing through the bend, will either get stuck or bend itself. The latter may damage it.

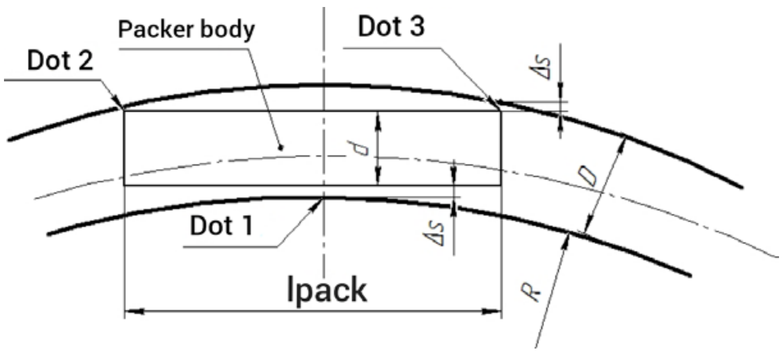


Figure 3. Scheme of the maximum free fit of the packer.

The friction force of the sealing element of the packer against the wall of the casing $F_{friction}$ is determined to obtain the condition of the packer tightness, as well as to obtain data on the compensation of the buoyancy force from the pressure drop due to friction forces:

$$F_{fric} = \mu_{fr} \cdot f_0 (S_1 + S_2) \quad (2)$$

where μ_{fr} - is the coefficient of sliding friction of rubber on steel, $\mu_{fr} = 0{,}6$;
f_0 – initial seal pressure, $f_0 = 101$ MPa;

S_1 – is the area of the inner side surface of the sealing element,

$$S_1 = 2\pi rh = 2\cdot 3{,}14\cdot 0{,}045\cdot 0{,}25 = 0{,}071\ m^2 \quad (3)$$

S_2 – the area of the outer side surface of the sealing element,

$$S_2 = 2\pi Rh = 2\cdot 3{,}14\cdot 0{,}1025\cdot 0{,}25 = 0{,}161\ m^2 \quad (4)$$

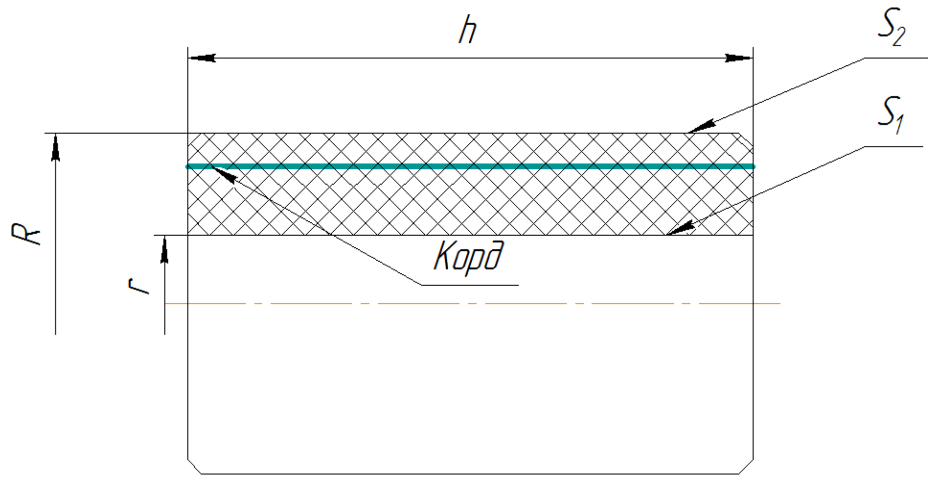


Figure 4. Sealing element diagram.

Friction force of the sealing element against the wall of the casing string:
$F_{fric} = \mu_{fr} \cdot f_0(S_1 + S_2) = 0{,}6 \cdot 101 \cdot 10^6 \cdot (0{,}071 + 0{,}161) = 14$ MN.
When $F_{fr} = 320$ kN the condition of the force acting on the borehole wall is fulfilled. Since $F_{fr} \geq [\, F_{fr}\,]$, then the tightness of the packer will be carried out.

Findings: In the packer, the contact pressure necessary for sealing between the well wall and the sealing rubber elements was determined, the axial force necessary for a reliable fit of the packer in the well was calculated, and the dimensions of the cord casing of the sealing element were calculated.

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ИССЛЕДОВАНИЕ ЭФФЕКТИВНОСТИ ГИДРОМЕХАНИЧЕСКИХ ПАКЕРОВ, ПРИМЕНЯЕМЫХ ПРИ ЭКСПЛУАТАЦИИ НАКЛОННЫХ СКВАЖИН

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РЕЗЮМЕ

В статье рассмотрена общие недостатки пакеров, предназначенных для проведения геолого – технических мероприятий в необсаженной скважине. Эти недостаткипакеров могут стать причиной возникновения осложнений в работе скважин, что ведет к увеличению непроизводительного времени или даже к потери скважины и как следствие приводит к большим затратам нефтегазодобывающих предприятий. Изучена конструкция раздвижных опорных элементов, центрирующих конструкцию в открытом стволе, предотвращая затекание уплотнительного резинового элемента вследствие возникновения перепада давлений, и обеспечивая в дальнейшем надежный срыв пакера.

Ключевые слова: гидромеханический пакер, уплотнительный элемент, ствол скважины, устройство пакера.

MAİLİ YÖNƏLDİLMİŞ QUYULARIN İSTİSMARINDA İSTİFADƏ OLUNAN HİDROMEXANİKİ PAKERİN İŞQABİLİYYƏTLİLİYİNİN TƏDQİQİ

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XÜLASƏ

Məqalədə quyu lüləsində geoloji – texniki tədbirlərin həyata keçirilməsi üçün nəzərdə tutulan pakerlərin ümumi çatışmazlıqlarına baxılır. Pakerlərin bu çatışmazlıqları quyuların istismarında problemlər yarada bilər ki, bu da qeyri-istehsal vaxtının artmasına və hətta quyunun itirilməsinə səbəb olur və nəticədə neft – qaz şirkətləri üçün yüksək xərclərə gətirib çıxarır. Sürüşən dayaq elementlərinin konstruksiyası, konstruksiyanı açıq quyuda mərkəzləşdirmək, təzyiq düşməsinin baş verməsi səbəbindən kipləşdirici rezin elementin sızmasının qarşısını almaq və sonradan pakerin etibarlı oturdulmasını təmin etmək tədqiq edilmişdir.
Açar sözlər: hidromexaniki paker, kipləndirici element, quyu lüləsi, paker qurğusu.

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APPLICATIONS OF POLYMER COMPOSITE MATERIALS IN OIL MACHINE BUILDING

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ABSTRACT

Among the measures taken to improve the efficiency and durability of oilfield equipment, devices and technical means, as well as pipeline systems operated in aggressive conditions, it is very important to use polymer materials and compositions based on them.
The paper outlines the basic principles of creating polymer composite materials with specified properties for oilfield equipment
Keywords: Polymer composite materials, oilfield equipment, aggressive environment, strength properties, corrosion, pipeline.

Polymer composite materials in recent years have been increasingly used in various industries, including oilfield engineering. The simplicity and accessibility of the method of obtaining these materials, a wide range of regulation of their properties, the mobility of composition changes and the use of local raw materials make them more attractive for the conditions of our republic.
The properties of composite materials based on modified polymers make it possible to recommend them for use in the structures of oil and gas field equipment. The authors [1-4] have shown that as a result of modification, it is possible to improve the manufacturability of polymer compositions, increase the adhesive strength, oil, gasoline resistance, resistance to gorenje, tribological and other properties.
The analysis of the state of the issue of the influence of oilfield aggressive media on the change in the main characteristics of polymer coatings, as well as the study of the durability of equipment when using coatings of various compositions showed that [5, 6]:
-this problem, taking into account the above, is very relevant;
-the change in the strength of deformable materials under the influence of media is not a random factor, characteristic only for a certain type of polymer materials. This change is caused by the physicochemical processes of interaction of the medium with the polymer, developing both on the surface of the solid and in its volume;
-the change in the strength properties of polymer composite materials under the influence of various media determines the need to develop methods for obtaining such materials and products from them that would have high strength and durability, as well as resistance to aggressive media;
-it is necessary to expand research in the direction of creating polymer compositions with predetermined properties;
-to deepen the field of knowledge on the study of the durability of the polymer material itself and compositions based on them under the influence of aggressive environments.

The solution of these problems is currently becoming one of the most important problems of polymer mechanics, polymer materials science and processing technology.
The analysis of foreign and domestic developments in this direction has shown that the main research in the field of polymer protective composite materials is carried out using functional oligomers and high-molecular polymers. Mainly epoxy, epoxy and bitumen containing elastomeric compositions have been developed. The most promising among them are works on the creation of compositions based on mixtures of functional oligomers, including epoxy resins with various elastomers. Such coatings have a number of advantages (high adhesion; strength characteristics, increased resistance to aggressive environments) compared to other polymer materials. However, these protective compositions also have the disadvantage of creating special conditions for their application and curing. There are alternative works where it is shown that the introduction of high-molecular compounds, including elastomers, into the composition of epoxy compositions makes it possible to obtain compositions based on these mixtures that have moisture resistance, resistance to thermal effects, and a low modulus of elasticity.
In recent years, polymer composite materials based on an epoxy oligomer, polybutadiene and a block copolymer of butadiene with styrene have been widely used in practice. To increase the hardness and abrasion resistance of the coating, fillers are introduced into its composition - chromium and aluminum oxide. Coatings of this kind are characterized by high cohesive strength and tensile strength. In addition, the profitability of styrene-butadiene rubber (BSK) elastomer is also explained by the fact that by changing the ratio of butadiene and styrene, the method of synthesis of rubber, a wide range of BSK with different physical and mechanical characteristics can be obtained.
Thermoplastic, thermosetting, friction, antifriction and fire-resistant composite materials used in various conditions have been obtained on the basis of elastomer-containing epoxy mixtures.
By introducing polyamide modified with liquid epoxidized cispolybutadiene into a mixture based on unsaturated rubber, a mixture of low molecular weight elastomers with epoxy groups or an oligomeric carboxylate elastomer and an epoxy oligomer, elastic wear-resistant coatings with high elasticity, heat resistance, etc. were obtained.
In addition, an increase in the strength and operational characteristics of these compositions based on liquid rubbers is achieved by introducing crushed vulcanizate into their composition. To increase the hardness of these compositions, epoxidized oligomers of aniline, phenol and formaldehyde are introduced into their composition.
It is known that an important characteristic of coatings is their adhesive characteristics. In elastomeric compositions, this characteristic can be regulated by the introduction of an epoxy oligomer and its derivatives into their composition.
Another distinctive characteristic of coatings based on polymer compositions is their thermo-oxidative stability. Various epoxidized oligomers play the role of stabilizer in the above compositions. This is of particular importance in the development of liquid composite materials. The anticorrosive liquid coating is obtained on the basis of a film-forming binder of acrylic, alkyd or epoxy resin, polyvinyl acetate or chlorinated elastomer, as well as a corrosion inhibitor. A number of liquid composite coatings have been obtained on the basis of latex or modified elastomer emulsions. Liquid oligomers with terminal amino groups are used in the development of coatings for application to dry or wet surfaces, as well as cured under water.

To protect steel pipelines from corrosion and mechanical damage during transportation and installation, a two-layer material is proposed-a melt of an epoxy composition and a ribbed layer of extruded polyethylene. A large number of compositions of protective coatings based on mixtures of various types of phenol-formaldehyde and epoxy oligomers have been proposed.
A corrosion-chemical resistant coating with high bending resistance and impact resistance is obtained on the basis of a new-milk phenol-formaldehyde oligomer.
With the use of bitumen of various compositions, waterproofing coatings containing rubber, metal powders and anticorrosive additives were obtained.
Modified with a non-polar elastomer, bitumen-polymer compositions were used as a multilayer hydrothermal insulation coating to protect underground pipelines.
To protect metal pipelines and their elements from corrosion, a two-layer anticorrosive sealing tape based on butyl rubber, vulcanizing agents, magnesium oxide and graphite as a producing layer based on polyvinyl chloride has been developed.
The composition with increased abrasion and heat resistance, impact strength and increased flammability is obtained on the basis of a mixture of carboxynitrile rubber. It is shown that on the basis of various elastomers it is possible to obtain protective coatings with good resistance to chemicals, water, weather resistance.
To cover the inner surface of the pipes of treatment facilities, a composition is proposed based on a solution of a mixture of butyl and chlorinated rubber in toluene, which has acid, alkali resistance, adhesion to metal and resistance to impact.
The use of polymer composite materials with specified properties for oilfield and petrochemical equipment is largely determined by the specificity of their operating conditions. As you know, along with the aggressiveness of the environment, temperature and pressure (stresses), structural factors have a huge impact on the corrosion of parts.
As a result of the occurrence of thermal stress in the welding sites and changes in the chemical composition of the metal due to strong heating during welding, the corrosion resistance of the equipment decreases. The same pattern is observed in places where the flange is welded. The material of the main equipment and its parts during welding should be the same in characteristics. Only under these conditions is the durability and stability of the equipment ensured in aggressive conditions. These factors should also be taken into account when artificially protecting equipment from corrosion.
It is known that the equipment of the oil and petrochemical industry in most cases is operated under the simultaneous influence of aggressive media: inorganic and organic acids, alkalis, solvents, salts, dry and important gases, etc. Different concentrations of these media, temperature, overpressure, friction, vibration, etc. together dramatically increase the rate of corrosion of equipment.
It should be taken into account that, unlike metals, non-metallic materials, especially of organic origin, which are polymer materials, significantly change their properties under the influence of these factors. Under the influence of temperature, load or aggressive environment, unlike metal, the strength characteristics, adhesive strength, chemical resistance and thermal stability of polymer coatings deteriorate sharply. Another feature is that when mating with a metal surface, due to different values of the temperature coefficient of linear expansion of the metal and the polymer material, shrinkage and delamination of the coating occurs. A high rate of destruction of the coating at elevated temperatures and stresses is

observed in the places of welds and stress accumulation of the main structure. These factors have a significant impact on the performance of structures.

In this regard, the following requirements are imposed on polymer protective materials: chemical resistance in environments containing solvents, organic and inorganic acids, salts, alkalis, etc.; mechanical strength; a certain ratio of thermal expansion coefficients of the film and the protected material; thermal stability and maximum stability of physical and mechanical properties within the temperatures regulated for the operation of the equipment, according to the technological regulations; heat resistance and thermal conductivity of the protective layer; high adhesion to substrates; high impermeability; the ability to glue by affordable, simple means; cheapness and accessibility of components.

In the light of these requirements, it is necessary to establish the nature of the destruction of protective coatings from various polymer coatings. Unfortunately, in modern operating conditions of oil and petrochemical equipment, due to the complex of environmental influences, several types of coating destruction occur simultaneously: thermal, photochemical, mechanical, biological, etc.

As a result of exposure to various reagents, heating and cooling, radiation, meteorological, geological and mechanical conditions, the above types of destruction occur in polymer coatings.

It should be taken into account that the nature of, for example, thermal degradation depends mainly on the chemical state of polymers. Polymers containing reactive groups have high adhesive strength to metal and, depending on the nature of the functional group -OH, -C1, -COON, etc., they are destroyed slowly during thermal exposure. This fact is taken into account when creating compositions for protective coatings.

The most characteristic for the operation of equipment in conditions of offshore oil production and transportation of oil on polymer coatings is also a biological effect - the action of various microorganisms, mold. In this regard, the coatings must be I filled with various metal oxides, asbestos, etc.

The destruction of the coating under the influence of liquid and gaseous aggressive media occurs as a result of the diffusion of reagents, their sorption, chemical transformation and diffusion of reaction products into the interfacial space. Naturally, the chemical stability of the coating under these conditions will depend on the structure of the polymer, its polarity, etc.

It is known that epoxy compounds have chemical resistance to various aggressive media and elevated temperatures, high strength and adhesive properties. The main disadvantage of epoxy compositions is their fragility and low impact strength. The decrease in strength can be accelerated in the presence of internal stresses in the polymer, which are formed during formation and uneven cooling. There are cases in which cracking occurs only from internal stresses.

Thus, the analysis of the choice of polymer composite materials with specified properties for oilfield equipment and their operational features allow us to draw the following conclusion:

-the experimental work carried out and the analysis of patent and scientific and technical literature confirmed the relevance and expediency of research in the development of polymer composite materials based on the elastomer-epoxy resin system for obtaining anticorrosive protective coatings;

-when developing and applying protective coatings, it is necessary to solve a complex of issues related to improving the technology of obtaining and applying parts to the surface, operational properties and reliability of structures;
-when developing anticorrosive coatings for oilfield and petrochemical equipment, it is necessary to take into account the specificity of their operating conditions.

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POLİMER KOMPOZİSİYA MATERİALLARININ NEFT MAŞINQAYIRMASINDA TƏTBİQİ

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XÜLASƏ

Neft mədən avadanlıqları, qurğular və texniki vasitələrin, habelə aqressiv şəraitdə istismar olunan boru kəmərləri sistemlərinin iş qabiliyyətinin və uzunömürlülüyünün artırılması istiqamətində görülən tədbirlər sırasında polimer materiallardan və onların əsasında hazırlanan kompozisiyalardan istifadə olunması çox vacibdir.
Məqalədə neft-mədən avadanlığı üçün verilmiş xassələrə malik polimer kompozisiya materiallarının yaradılmasının əsas prinsipləri təsvir olunur
Açar sözlər: polimer kompozisiya materialları, neft-mədən avadanlıqları, aqressiv mühit, möhkəmlik xüsusiyyətləri, korroziya, boru kəməri.

ПРИМЕНЕНИЕ ПОЛИМЕРНЫХ КОМПОЗИЦИОННЫХ МАТЕРИАЛОВ В НЕФТЯНОМ МАШИНОСТРОЕНИИ

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РЕЗЮМЕ

Использование полимерных материалов и композиций на их основе занимает важное место среди мероприятий, направленных на повышение эффективности и долговечности работы нефтепромыслового оборудования, сооружений и технических средств, а также трубопроводных систем, работающих в агрессивных средах.
В статье описаны основные принципы создания полимерных композиционных материалов с заданными свойствами для нефтепромыслового оборудования.
Ключевые слова: полимерные композиционные материалы, нефтепромысловое оборудование, агрессивная среда, прочностные характеристики, коррозия, трубопровод.

EQUIPMENT. TECHNOLOGIES. MATERIALS

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METHOD OF IMPROVING THE COOLING SYSTEM IN THE SPINDLE SECTION OF TURBODRILLS

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ABSTRACT

The article proposes a new way to improve the cooling system of the spindle section of turbodrills. The constructive part, purpose and principle of operation of the spindle section of currently used turbodrills were explained and their shortcomings were shown. It was found that the number of parts in the design is many, there are four bearing rings in one bearing, this number should be reduced. The ball treadmill has many disadvantages; The rolling surfaces of bearing rings are convex and concave surfaces of truncated cones located under and above a row of balls, in which there is a method of point contact, i.e. Each bearing ring has one contact point (four in total). Also, the bearing cooling system has disadvantages, their design also needs to be changed. A new spindle design was proposed and all the shortcomings were eliminated. The number of bearing rings has decreased; A convenient and efficient treadmill is provided under and over the row of balls to roll the balls. In the form of concave and convex one and a half surfaces. The cooling system of the ball bearing set has been improved. An interesting trajectory of movement of a part of the drilling fluid in the form of a zigzag is proposed, where the correction of its movement changes several times, overcoming a large distance of movement and having more contact time with the bearing parts, cools them qualitatively and comprehensively.

Keywords : turbodrill, spindle section, cooling system, ball bearing, treadmill, flushing mud, spindle housing, shaft, thin ring.

Introduction: All over the world, countries are interested in drilling deep and ultra-deep wells. Each country has its own interest; some need production wells and some need research wells . However, they are united by the complexity of drilling such wells because the work is carried out in very hard rocks at high temperatures and using high-density drilling mud.

Currently, existing technologies do not allow for effective drilling in such conditions due to the fact that the loads on the drill string, bit, and hydraulic motor increase.

The most common downhole screw motors are unable to operate under these conditions due to the significant temperature limitation (up to 150 ◦C). In addition, their indisputable advantages in the field of creating significant torque elements at low speeds lead to the inappropriateness of their use in these conditions.

Based on the foregoing, the most effective technology for drilling such wells is the use of impregnated bits in combination with a high-speed hydraulic turbine engine.

The first turbine apparatus for drilling wells was developed in 1923 by M.A. Kapelyushnikov, together with S.M. Volokh. and Kornev N.A., called the Kapelyushnikov turbodrill (12 HP,

single-stage turbine, multi-stage planetary gearbox). The final design of the turbodrill, which has become widespread, was created by PP Shumilov, EI Tagiyev, and MT Gusman .
At the moment, there are high-speed turbodrills in Russia, the rotational speed of which does not exceed 1300 rpm (revolutions per minute), for example, turbodrills developed by VNIIBT-Drilling Instrument. These include models of turbodrills T-4 3/4, T-6 3/4, and TSSh-178T.
Abroad, there are turbodrills whose rotational speed reaches 2500 rpm, for example, Neyrfor TTT 2 1/8 and Neyrfor TTT 2 7/8 , developed by Schlumberger. However, these turbodrills have a significant limitation associated with their diameter, hence the small working area of the blades, due to which the hydraulic energy of the drilling fluid flow is converted into the rotation of the turbodrill shaft.
From the above technical limitation, it follows that it is necessary to develop a turbodrill with a rotation speed, troubleshooting with mechanical parts and improve them, reducing the drilling time and reducing economic costs, while maintaining high reliability.

Theory basis: Downhole motors used to rotate the drill bit in drilling oil and gas wells. Their spindle section consists of multi-row ball-shaped radial-support separator-free bearings; the number of rows is 10 or 12; the number and diameter of balls in each row depends on the diameter of the turbodrill. Currently, spindle turbochargers used in Azerbaijan are produced at the bearing factory in Minsk, Belarus.

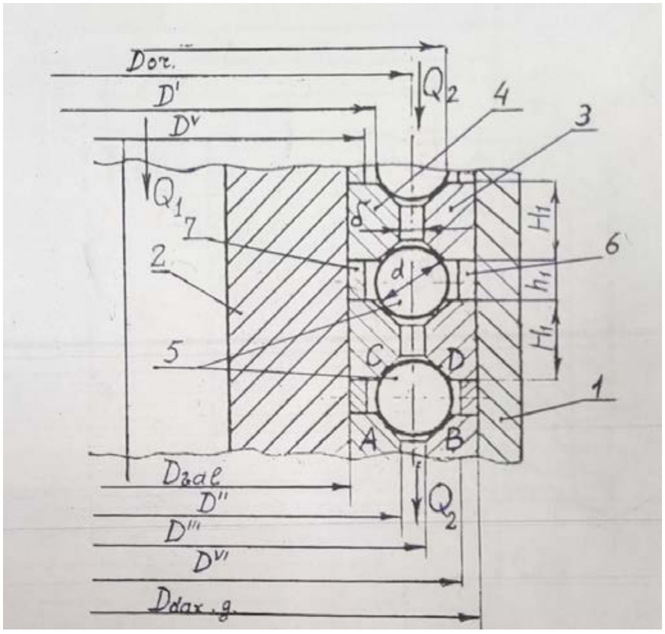


Figure 1. Drawing of a part of the spindle is presented.

The spindle section consists of the following parts: 1 - spindle body; 2 - spindle shaft; 3 - stator of the bearing; 4 - the rotor of the bearing; 5 - balls; 6 - large diameter thin ring; 7 - small diameter thin ring.

The spindle section of the turbine produced by the Minsk plant must be installed as follows. A row of balls is arranged with two bearing rings at the bottom and two bearing rings at the top, i.e four bearing rings; two of the rings are the stator of the bearing, one below the row, another above the row and located tightly on the inner surface of the spindle body; the other two rings are the rotor of the bearing, one at the bottom of the row and the other at the top and mounted on the outer surface of the spindle shaft. One ball bearing also has two thin rings, one large in diameter and the other small in diameter; the large diameter balls are clamped to the inner surface of the spindle body outside the row of balls, and the small diameter balls are placed inside the row of balls and clamped to the outer surface of the spindle shaft. These thin rings keep the lower and upper bearing rings pressed together when the turbodrill is running, preventing them from moving parallel to the turbodrill axis, so such displacement can occur when the bearing rings are worn.

In this spindle design, the escape path of the balls is created as follows: A concave conical surface is formed inside the stator ring under the balls of a bearing, and a convex conical surface is formed outside the rotor ring, thus obtaining a runway under the row of balls; inside the stator ring located on the balls of the same bearing, a convex conical surface is formed, A convex cross-sectional conical surface is created outside (at the bottom) of the rotor ring, thus creating a runway over the row of bearing balls. In the static position of the turbodrill, each ball has four points of contact with the bearing rings (there are four rings) (one point of contact in each bearing ring).

During the drilling of the well, the turbodrill operates and the spindle section is in a dynamic position, i.e the rotor rotates, the stator remains at rest, the balls rotate in four circles and new contact point occurs.

It is known that when a well is drilled, the drill ax is loaded with tons of length and the load on each ball is high at point contact, the balls and bearing rings are worn out, the backlash size increases and when the backlash reaches 10 mm the turbodrill is lifted from the well and sent to the pipe base for repair.

The above is shown that the structure used has defects; It should be mentioned that the spindle section currently used in Azerbaijan is also defective.

When inclined and bent wells are drilled with a turbodrill, radial displacements occur in its rotor; in this case, the annular space between the stator and rotor rings expands on one side and narrows on the other and even the rotor rings rub against the stator rings and they are worn out, the backlash grows and the turbodrill loses its ability to work; also, here in the event of a vertical displacement, the balls roll over the sharp edge of the stator's outer circumference and they wear out and destroy it. This is one of the shortcomings of the old turbodrills.

During the drilling process, the flushing liquid injected into the well drilled by the drilling pump passes through the turbine section of the turbine and is divided into two parts when it reaches the spindle section; main part of solvent Q_1 descends from the inside of the spindle shaft, other part Q_2 is given to the set of bearing is used for cooling. With Q_2 flow the drilling fluid is fed into the annular space between the bearing rings of the spindle rotor and the stators, moves down with a vertical gap. In such cooling system, only the balls are normally cooled down, and the bearing rings are not cooled completely, because the fluid collected on the right and left of the balls remains motionless. This cooling system should be upgraded. A new design of such a spindle was created and a patent of the Republic of Azerbaijan was obtained [7].

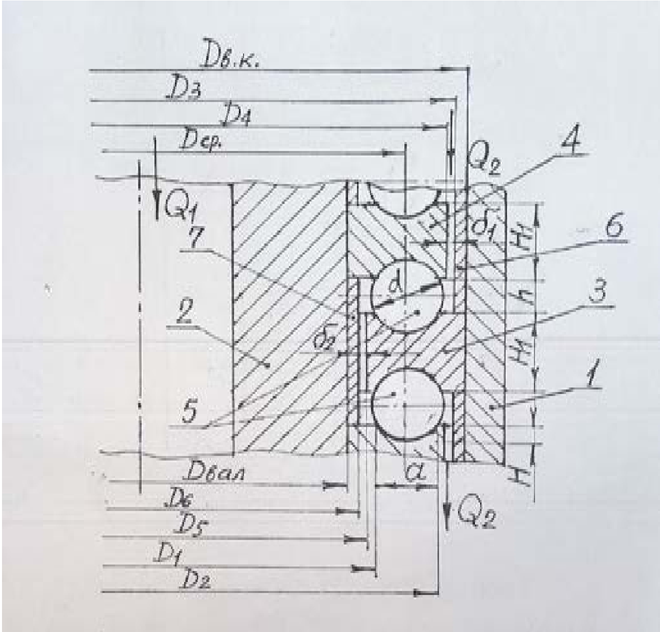


Figure 2. Drawing of a part of the new constructed spindle is presented.

A cross-sectional drawing of a part of the newly constructed spindle section proposed for application in A7Ш type turbo drill is presented. This spindle consists of the following parts: 1 – spindle body; 2 - spindle shaft; 3 - stator of the bearing; 4 - rotor ring of the bearing; 5 - balls; 6 - large diameter thin ring; 7 - thin ring with small diameter.

The movement of the cooling part of the drilling mud with a set of bearings (new design) has a very interesting trajectory. To better understand it, the newly proposed construction of the spindle is first explained below.
A ball bearing has two bearing rings, one of which is the stator ring, which is fitted to the inner surface of the spindle body, the other bearing ring is a rotor, which sits snugly on the outer surface of the spindle shaft. A large-diameter thin ring compresses adjacent stators to form a seal. A thin ring of small diameter compresses the adjacent rotor bearing rings and creates a seal. When the turbocharger is not working, ie in a static position, the ball rings come into contact with the balls along a line (ie a circular arc), which is called linear contact.
In the linear contact method, the loads acting on the spindle details are relatively reduced and their longevity and reliability are increased.
Flushing liquid flows down with a vertical annular gap between spindle first rotor bearing ring and large diametr thin ring with Q_2 flow, then centrifugal flat-radial flow moves up to a small diameter thin ring in the space between the row of balls between the rotor and stator bearing rings; then the washing solution moves down through the vertical annular gap between the stator ring of the cushion and the thin ring of small diameter; then the drilling fluid moves from the small-diameter annular cavity to the second large-diameter thin ring with a centrifugal flat-radial flow in the annular space between the stator of the bearing and the rotor ring of the lower second bearing (where the balls of the second bearing are located). Thereby,

the cooling process of the newly constructed bearing takes place and all 10 ÷ 12 bearings which are placed under each other are cooled in sequence. The advantages of this proposed new cooling system are:

The length of the distance which the coolant liquid passes on the new designed spindle:

$$L_1=2H_1+2(D_3-D_6) \tag{1}$$

In an old-design spindle, the length of the path through which the coolant liquid passes:

$$L_2=2H_1 \tag{2}$$

There is difference:

$$L_1\text{-}L_2=2(D_3-D_6) \tag{3}$$

As you can see $L_1 > L_2$ –dir.

On the other hand, in the new design, the solution has a long contact time with the details of the spindle, and the details are cooled on all sides, which is the second advantage of the new cooling system.

Conclusions: The article proposes a new method of improving the cooling system in the spindle section of turbodrills.

1. The constructive part, purpose and principle of operation of the spindle section of currently used turbodrills were explained and their shortcomings were shown.
2. It has been determined that the number of parts involved in the old construction is large, there are 4 rings in one bearing and this number should be reduced.
3. The ball treadmill has many disadvantages; The rolling surfaces of bearing rings are convex and concave surfaces of truncated cones located under and above a row of balls, in which there is a method of point contact, i.e. Each bearing ring has one contact point (four in total).
4. The bearing cooling system has disadvantages, their design also needs to be changed.
5. A new spindle design was proposed and all the shortcomings were eliminated.
6. The number of bearing rings has decreased 2 times.
7. A comfortable and efficient escape route is offered, with concave and convex semicircular surfaces under and above the row of balls.

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СПОСОБ УСОВЕРШЕНСТВОВАНИЯ СИСТЕМЫ ОХЛАЖДЕНИЯ ШПИНДЕЛЬНОЙ ЧАСТИ ТУРБИН.

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РЕЗЮМЕ

В статье предложен усовершенствованный новый способ охлаждения шпиндельной части турбокомпрессоров. Объяснены конструкция, назначение и принцип работы используемой в настоящее время секции шпинделя турбины, показаны дефекты. Выяснилось, что количество деталей в конструкции большое, в одной подушке 4 кольца подушки; необходимо уменьшить это число. Путь побега шаров очень несовершенен; Поверхности качения опорных колец состоят из выпуклых и вогнутых конических поверхностей под и над шариками, которые имеют точечный метод контакта, т.е. каждое опорное кольцо имеет одну точку контакта, всего четыре точки. Система охлаждения комплекта подушек шпинделя также неисправна; его конструкция должна быть изменена. Предложена новая конструкция шпинделя и устранены все эти недостатки. Количество колец для подушек уменьшено вдвое. Был предложен удобный и эффективный путь эвакуации; ниже и выше ряда шаров образуются вогнутые и выпуклые полукруглые поверхности. Усовершенствована система охлаждения комплекта подушек шпинделя. Предложена интересная зигзагообразная траектория движения охлаждающей части промывочной буровой жидкости, причем все детали подушки качественно охлаждаются со всех сторон.

Ключевые слова: турбобур, шпиндельная секция, система охлаждения, беговая дорожка, промывочный буровой раствор, вертикальное движение, ряд шаров, корпус шпинделя, вал.

EQUIPMENT. TECHNOLOGIES. MATERIALS

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TURBOBURLARIN ŞPİNDEL SEKSİYASINDA SOYUTMA SİSTEMİNİN TƏKMİLLƏŞDİRİLMƏSİ ÜSULU

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XÜLASƏ

Məqalədə, turboburların şpindel seksiyasında soyutma sisteminin təkmilləşdirilmiş yeni üsulu təklif olunmuşdur. Hal-hazırda tətbiq edilən turboburun şpindel seksiyasının konstruksiya quruluşu, təyinatı və və iş prinsipi izah edilmişdir və qüsurları göstərilmişdir. Aşkar olunmuşdur ki, konstruksiyada olan detalların sayı çoxdur, bir yastıqda 4 yastıq həlqəsi vardır; bu sayı azaltmaq lazımdır. Kürəciklərin qaçış yolu çox qüsurludur; yastıq həlqələrinin diyirlənmə səthləri kürəciklərin altında və üstündə qabarıq və çökək kəsik konuslar səthlərindən ibarətdirlər, bunlarda nöqtəvi təmas üsulu mövcuddur, yəni hər yastıq həlqəsində bir təmas nöqtəsi, cəmi dörd nöqtə vardır. Şpindeldəki yastıqlar dəstinin soyudulma sistemi də qüsurludur; onun konstruksiyası dəyişdirilməlidir. Şpindelin yeni konstruksiyası təklif edilmişdir və bu qüsurların hamısı aradan qaldırılmışdır. Yastıq həlqələrinin sayı iki dəfə azaldılmışdır. Rahat və səmərəli qaçış yolu təklif edilmişdir; kürəciklər cərgəsinin altında və üstündə çökək və qabarıq yarımtor səthləri yaradılmışdır. Şpindeldəki yastıqlar dəstinin soyudulma sistem təkmilləşdirilmişdir. Yuma qazma məhlulunun soyuducu hissəsinin hərəkətinə ziqzaq şəkilli maraqlı hərəkət trayektoriyası təklif edilmişdir və yastığın bütün detalları hər tərəfdən keyfiyyətlə soyudulur.

Açar sözlər: turbobur, şpindel seksiyası, soyutma sistemi, qaçış yolu, qazma yuma məhlulu, vertikal hərəkət, kürəciklər cərgəsi, şpindelin gövdəsi, val.

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CONSTRUCTION OF INDICATOR DIAGRAM OF A DRAWER DEEP PUMP OIL WELL PRODUCING VISCOUS-PLASTIC OIL AND SURROUNDED WITH THICKENED WATER

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ABSTRACT

In the article, indicator diagrams of a sucker-rod well pumping well producing viscous-plastic oil and watered with thickened water are constructed. The well was investigated by a hydrodynamic method under stationary inflow conditions and all the necessary formulas were derived to determine the nature of the filtering fluids in the reservoir. There are six different formulas for reservoir pressure.
Keywords: sucker rod well, viscous-plastic oil, thickened water, indicator diagram, bottomhole static limiting pressure, bottomhole static limiting pressure, reservoir pressure, initial pressure drop, initial pressure gradient.

The relevance of researchж For viscous-plastic oil and thickened reservoir water, the first indicator diagrams were built, which were extended until they intersected with the bottomhole pressure axis; the abscissas of the intersection points give the lower static bottomhole pressure for oil $P_H^I$and the lower static bottomhole pressure for thickened water $P_в^I$; as you can see: $P_в^I$> P_H^I. Further, the pumping unit pumps the estimated volume of liquid into the reservoir through the working well; fluid injection is stopped and the echometer of the "Hardware and software complex Quantor-4 micro" records the drop curve of the thickened water level depth in the well, and the ordinate of the stabilized part of this curve is the upper static limiting bottomhole pressure of the thickened water $P_в^{II}$. The resulting difference $P_в^{II}$is $P_в^I$equated to 2Δ $P_{0в}$, where Δ is $P_{0в}$the initial pressure drop for thickened water; half of this difference helps to find the reservoir pressure value. From point O on the right, segment Δ $P_{0H}$is plotted and point B is located on the bottomhole pressure axis; the abscissa of point B gives the upper limit of static bottom hole pressure for oil P_{H}^{II}. Further, the first indicator diagram is extended in its direction until it intersects with the viscous-plastic oil flow rate axis (with the ordinate axis) and the value of the potential oil flow rate is found, which makes it possible to judge the possibility of increasing the oil flow rate (or there is no such possibility).

Methods: It is known that in order to select the correct optimal technological mode of operation of oil and gas wells, the well is examined by a hydrodynamic method at steady-state inflow modes and indicator diagrams are built: two diagrams for oil and two diagrams for thickened water. This research method is widely used in oil and gas fields, however, in sucker rod wells producing viscous-plastic oil and flooded with thickened water, this method was not used and there is no information about it in the literature. In this article, for the first time, such a research method is proposed and applied.

In the proposed method, research operations are performed with the following sequence.
First, the depth of the face is measured with a load; if a sand plug is found, then it is washed and the bottom is cleaned; oil and water flow rates are measured, as well as dynamic bottomhole pressure in its mode; with these data, the first indicator charts for oil and water are built.
Figure 1 shows indicator diagrams for a well with marked features.
The first indicator chart for oil is extended in its direction, intersects with the bottomhole pressure axis (with the abscissa axis) and the lower static bottom hole pressure limit for oil is obtained P_{H}^{I}. The first indicator diagram for gelled water is extended in its direction, intersects with the axis of bottomhole pressures and the lower static limiting bottomhole pressure for thickened water is obtained P_{B}^{I}. The slope of this diagram relative to the pressure axis is greater than the slope of the diagram for viscous-plastic oil, therefore, the initial pressure drop for thickened water ΔP_{s} is less than for viscous-plastic oil.
The next research operations are as follows. From the annulus of a working well, a pumping unit pumps viscous-plastic oil and thickened water into the productive formation at constant rates; stop pumping and take a pressure drop curve for thickened water; point A and value are found $2\Delta P_{OH}$ along the ordinate of the stabilized final part of the curve ; from the midpoint of this value, a segment is laid on the right $2\Delta P_{OH}$; and find point B and get $2\Delta P_{OH}$. The abscissa of point A gives the upper static bottomhole pressure limit P_{B}^{II} for thickened water; the abscissa of point B gives the upper static bottomhole pressure limit P_{H}^{II} for viscous oil; the abscissa of point O gives the reservoir pressure value P_{K}.

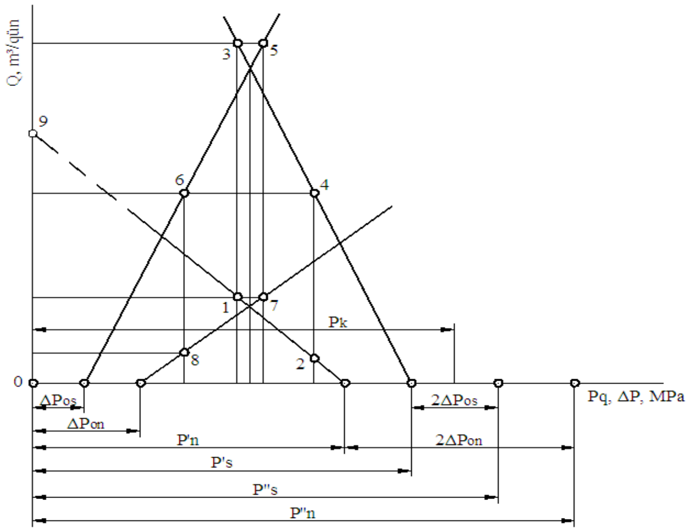


Figure 1. Indicator diagrams of a sucker-rod pumping well producing viscous-plastic oil and thickened water.

Using the obtained research data, the following calculations of the quantities were carried out:
Initial pressure drop for oil:

$$\Delta P_{OH} = \frac{1}{2}\left(P_{H}^{II} - P_{H}^{I}\right) \quad (1)$$

Initial pressure drop for thickened water:

$$\Delta P_{OB} = \frac{1}{2}\left(\mathrm{P}_{\text{в}}^{II} - \mathrm{P}_{\text{в}}^{I}\right), \tag{2}$$

For reservoir pressure:

$$\mathrm{P}_{\text{К}} = \frac{1}{2}\left(\mathrm{P}_{H}^{I} - \mathrm{P}_{H}^{II}\right) = \frac{1}{2}\left(\mathrm{P}_{\text{в}}^{I} - \mathrm{P}_{\text{в}}^{II}\right) = \mathrm{P}_{H}^{I} + \Delta P_{OH} = \mathrm{P}_{H}^{II} - \Delta P_{OH} = \mathrm{P}_{\text{в}}^{I} + \Delta P_{OB} = \mathrm{P}_{\text{в}}^{II} - \Delta P_{OB} \tag{3}$$

six formulas are derived to determine the value of reservoir pressure .$\mathrm{P}_{\text{К}}$
The initial pressure gradient that appears for a viscous-plastic oil will be:

$$G_H = \frac{\Delta P_{OH}}{R_K - r_C}, \tag{4}$$

The initial pressure gradient that appears for thickened water will be:

$$G_{\text{в}} = \frac{\Delta P_{O\text{в}}}{R_K - r_C}, \tag{5}$$

If the first indicator chart for oil is extended in its direction to intersect with the y-axis (with the rate axis), then the value of the potential oil rate is obtained, which indicates whether there is an opportunity to increase the oil rate or not; if so, to what extent? Determining the potential oil production rate also makes it possible to establish the correct optimal operating mode for the well under study.
The next research operations are performed as follows. The abscissas of points 1 and 2 of the first indicator diagram for thickened water are subtracted from the reservoir pressure $\mathrm{P}_{\text{К}}$and the abscissas of points 3 and 4 are found, which are connected by a straight line and the second diagram for thickened water is obtained. This diagram intersects with the axis of depressions Δ P and a segment is obtained ΔP_{OB}. The abscissas of points 5 and 6 of the first indicator diagram for viscous-plastic oil are subtracted from the reservoir pressure $\mathrm{P}_{\text{К}}$and the abscissas of points 7 and 8 are found, which are connected by a straight line and the second indicator diagram for viscous-plastic oil is obtained. This diagram intersects with the axis of depressions Δ P and a segment is obtained ΔP_{OH}.
If the first diagram for viscous-plastic oil is extended in its direction towards the flow rate axis (y-axis), then intersecting with it gives the value of the potential flow rate of viscous-plastic oil, which allows you to check the presence or absence of the possibility of increasing the oil production rate.
On the second diagram, one point C is taken and the values of the main filtration parameters of viscous-plastic oil are determined from its coordinates; on the second diagram for thickened water, one point E is taken and the values of the main parameters of thickened water filtration in the reservoir are determined from its coordinates.

Findings: In the article, indicator diagrams of a sucker-rod pumping well producing viscous-plastic oil and flooded with thickened water are constructed.

1. The well was investigated by a hydrodynamic method in steady flow regimes, and all the necessary formulas were derived to reveal the nature of the filtration of these fluids from the formation to the well.
2. For reservoir pressure, six different formulas are defined.

3. For viscous-plastic oil and thickened water, the first indicator diagrams built were extended in their directions until they intersect with the axis of bottomhole pressures and the lower static limiting bottomhole pressures were found.
4. By means of a pumping unit, liquid was pumped into the reservoir in the calculated volume, the injection was stopped, and the bottomhole pressure drop curve was taken without delay.
5. On the ordinate of the stabilized final part of this curve, the point A and the value are found $2\Delta P_{OB}$, in the middle of this segment, the point O gives the value P_K.
6. The value $\Delta P_{OH}$is plotted on the right side from point O and point B is located, which gives the value of the upper static bottomhole pressure limit for viscous-plastic oil.
7. Using the obtained research data, calculations were carried out and all values were determined that reveals the nature of fluid filtration from the formation to the well.
8. Taking points C and E on the second diagrams, it is possible to determine the main parameters of the reservoir from their coordinates, i.e. coefficients of permeability, hydraulic conductivity, and non-conductivity and the reduced radius of the well.

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ŞTAQNLI DƏRİNLİK NASOS QATILAŞDIRILMIŞ SU İLƏ SULANMIŞ ÖZLÜ-PLASTİK NEFT VERƏN QUYUNUN İNDİKATOR DİAQRAMLARININ QURULMASI

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XÜLASƏ

Məqalədə, ştanqlı dərinlik nasos qatılaşdırılmış su ilə sulanmış, özlü-plastik neft verən quyunun indikator diaqramları qurulmuşdur. Quyu qərarlaşmış rejimlərində hidrodinamik üsulla tədqiq edilmişdir və lazım olan süzülmə parametrləri üçün düsturlar çıxarılmışdır. Lay təzyiqi üçün altı düstur verilmişdir.
Açar sözlər: ştanqlı dərinlik nasos quyusu, özlü-plastik neft, qatılaşdırılmış su, indikator diaqramı, aşağı statik limit quyudibi təzyiq, yuxarı statik limit quyudibi təzyiq, lay təzyiqi, başlanğıc təzyiq düşgüsü, başlanğıc təzyiq qradiyenti.

ПОСТРОЕНИЕ ИНДИКАТОРНЫХ ДИАГРАММ ШТАНГОВОЙ ГЛУБИННОНАСОСНОЙ НЕФТЯНОЙ СКВАЖИНЫ, ПРОДУЦИРУЮЩЕЙ ВЯЗКО-ПЛАСТИЧНОЙ НЕФТЬЮ И ОБВОДИВШЕЙСЯ ЗАГУЩЕННОЙ ВОДОЙ

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РЕЗЮМЕ

В статье построены индикаторные диаграммы штанговой глубинно-насосной скважины, продуцирующей вязко-пластичную нефть и обводненную загущенной водой. Скважина исследована гидродинамическим способом при стационарных режимах притока и выведены все необходимые формулы для определения характера фильтрующихся жидкостей в пласте. Для пластового давления представлены шесть различных формул.
Ключевые слова: штанговая глубинно-насосная скважина, вязко-пластичная нефть, загущенная вода, индикаторная диаграмма, нижнее статическое предельное забойное давление, верхнее статическое предельное забойное давление, пластовое давление, начальный перепад давления, начальный градиент давления.

EQUIPMENT. TECHNOLOGIES. MATERIALS

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COMPARATIVE STUDY OF THE RELIABILITY OF A CENTRIFUGAL PUMP

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ABSTRACT

In this study, a centrifugal pump was analyzed to examine the effect of the use of inductors on its performance. Tests have been carried out to obtain optimal hydraulic performance before and after using the inductor. We used two types of inductors (axial inductors, screw inductors). For this, a test setup with a pumping system was specially designed to investigate the parameters under consideration. In this study, a simulation of the performance of a centrifugal pump in combination with inductors using CFD is presented for comparison with experimentally observed values. The model explores the effect of using inductors on pump head and flow. The results of the CFD model and experiment correlate well. In addition, the results assist pump decision makers in future developments in pump performance by ensuring that the pumps are in a safe and reliable operating condition. It can also be used for a wide variety of high head and flow pump applications.
Keywords: hydraulics, Centrifugal pump, reliability, pumps, devices, system.

Introduction: The pump model combined with the inductor is an important part of this study. The CFD model examines the effect of using an inductor on head and flow curves. The result of this stage has a huge impact on the quality of the results obtained using the developed mathematical model and experimental setup. The main problem is to model the geometry of the inductor together with the pump. In fact, part of this study is only intended to ascertain the correlation between data generated by CFD models and experimentally observed data. If the CFD model is developed in such a way as to establish a good correlation with the observed data, it can be used to analyze various types of system without creating a physical model. From a purely numerical point of view, the geometric complexity of the problem lies in the fact that it requires the use of the full power of modern computational fluid dynamics to solve the equations of motion and turbulence models in areas that include several surfaces.

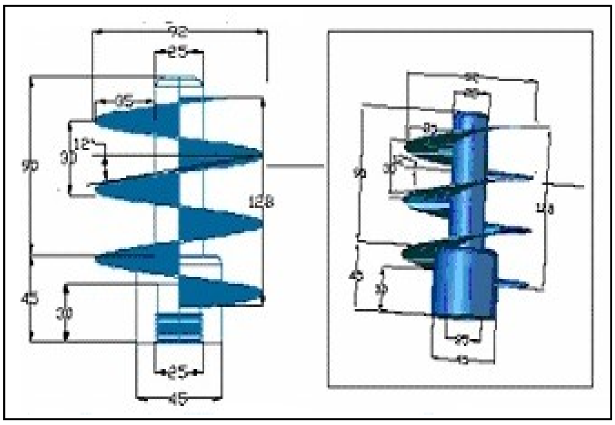


Figure 1: Design parameters of helical inducer.

Without helical and axial inducers.

Numerical simulation and model building Numerical simulation was performed after the completion of the experimental measurements. The simulation was performed using the CFD code, i.e., FLUENT software under ANSYS version 12.0, commercial code with 3D flow solution. The FLUENT model serves as a tool that discretizes and solves the governing equations for a particular geometry using a finite volume set (2).

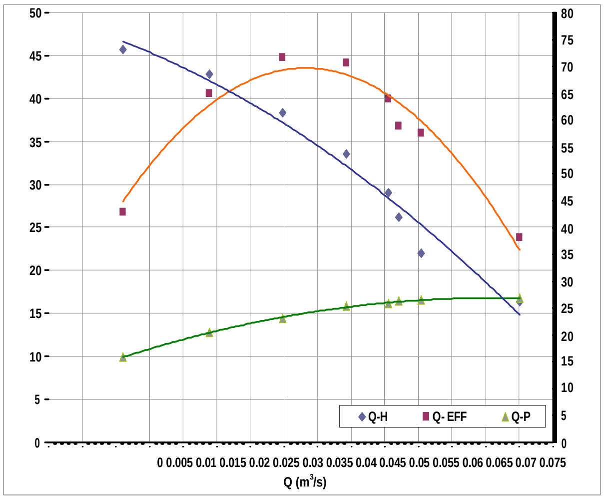


Figure 2. Pump performance curve using helical inducer.

Geometry design Basic 3D geometry is created using CFturbo. Generated mesh with GAMBIT. The meshes required for the calculations were generated and validated using the CFD pre-processing package, GAMBIT. Due to the very complex geometry of the pump, an unstructured tetrahedral mesh is used. "Equi-Angle Tilt" and The EquiSize Skew” meshes were less than 0.95, so the quality of the mesh is good. On fig.3shows a 3D model and mesh of the computational domain wall with blades.

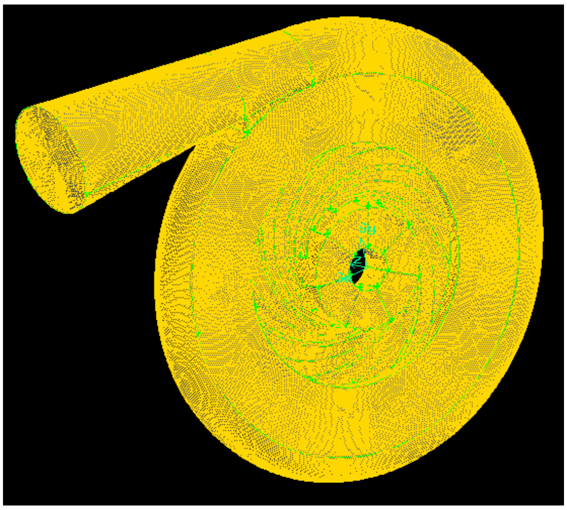


Figure 3. The pump 3D model and wall grid.

Numerical Simulation of the Pump Model the FLUENT program was used to simulate the internal flow field in the absence of cavitation. The standard k-ε turbulence model and the SIMPLEC algorithm are applied to solve the RANS equations. The simulation is stable and a moving frame of reference is used to account for the interaction between the impeller and the volute. The accuracy of the convergence of residues is 10-5. 4.4 Control Equations The flow of an incompressible fluid through a rotating impeller is solved in a moving reference frame with a constant rotation speed equal to the rotation speed of the impeller. The flow through the fixed parts of the pump is solved in an inertial frame of reference. The basic equations for the impeller are as follows:

$$A_{qu_r} = 0 \tag{1}$$

$$A_{qu_r} + 2q\Omega * u_r + q\Omega * \Omega * r = -AP + \mu_{\text{eff}} A^2 u_r \tag{2}$$

Where,

- **ρ**, is the density of the fluid
- **P** is the static pressure,
- **ur** is the vector fluid velocity in the rotating system,
- **Ω** is the rotational speed and μeff is the dynamic effective viscosity which is a linear combination of laminar and turbulent viscosity derived from k-ε model of turbulence.

The last two terms in the left-hand side of Eq. (2) are the effects of the Coriolis and centrifugal forces due to the rotating frame of reference. For the stationary parts of the centrifugal pump, the governing equations are formulated in the stationary reference frame. The continuity equation remains the same, but the momentum equation reduces to:

$$A_{qu} = -AP + \mu_{eff} A^2 u \tag{3}$$

where,

u - is the vector fluid velocity in the stationary frame of reference. The turbulence of the flow is modeled with standard
k-ε model that is rated as the most used model that combines simplicity, robustness and reasonable accuracy considered 100 µm. The turbulence intensity at the inlet totally depends on the upstream history of flow. Since the fluid in the suction tank is undisturbed, the turbulence intensity for all conditions is considered 5%. Water was used as a working fluid in ambient condition.

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MƏRKƏZDƏNQAÇMA NASOSUNUN ETİBARLILIĞININ MÜQAYİSƏLİ TƏDQİQİ

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XÜLASƏ

Bu işdə induktorların istifadəsinin mərkəzdənqaçma nasosuna təsirini təhlili aparılmışdır. İnduktordan istifadə edilməzdən əvvəl və sonra optimal hidravlik performansı əldə etmək üçün sınaqlar aparılmışdır. İki növ induktordan istifadə edilib (aksial induktorlar, vintli induktorlar)..Bu işdə eksperimental olaraq müşahidə edilən qiymətlərlə müqayisə üçün CFD istifadə edən induktorlarla birlikdə mərkəzdənqaçma nasosun işinin simulyasiyası təqdim edilmişdir. Model induktorlardan istifadənin nasosun başlığına və axınına təsirini araşdırır. CFD modeli və eksperimentin nəticələri yaxşı əlaqələndirilir. Alınan nəticələr nasosların təhlükəsiz və etibarlı işləmə vəziyyətində olmasını təmin edərək, həmçinin müxtəlif yüksək basqılı və məhsuldarlıqlı nasosların tətbiqləri üçün istifadə edilə bilər
Açar sözlər: hidravlika, mərkəzdənqaçma nasosu, etibarlılıq, nasoslar, qurğular, sistem.

EQUIPMENT. TECHNOLOGIES. MATERIALS

VOLUME 11 ISSUE 03 2022

СРАВНИТЕЛЬНОЕ ИССЛЕДОВАНИЕ ХАРАКТЕРИСТИК НАДЕЖНОСТИ ЦЕНТРОБЕЖНОГО НАСОСА

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РЕЗЮМЕ

В этом исследовании центробежный насос был проанализирован для изучения влияния использования индукторов на его производительность. Испытания были проведены для получения оптимальных гидравлических характеристик до и после использования индуктора. В работе использовались два типа индукторов (аксиальные индукторы, винтовые индукторы). В настоящем исследовании представлено моделирование производительности центробежного насоса в сочетании с индукторами с использованием CFD для сравнения с экспериментально наблюдаемыми значениями. Модель исследует влияние использования индукторов на напор и подачу насоса. Результаты модели CFD и эксперимента хорошо коррелируют. Кроме того, результаты помогают лицам, принимающим решения по насосам, для будущих разработок в области производительности насосов, обеспечивая безопасное и надежное рабочее состояние насосов. Также его можно использовать для большого количества применений насосов с большим напором и производительностью.
Ключевые слова: гидравлика, центробежный насос, надежность, насосы, устройства, система.

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RESEARCH OF INCREASING THE CAPACITY OF THE MAIN ELEMENTS OF A SUBMERSIBLE CENTRIFUGAL ELECTRIC PUMP DEVICE

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ABSTRACT

The article is devoted to the investigation of the causes of failures in centrifugal submersible electric pumping units (CSEP) during the work process, the measures taken to eliminate failures, improve performance and increase the time between repairs. In order to improve the performance of CSEP, a number of changes were made in the design.

Keywords: centrifugal submersible electric pumping unit, guidance apparatus, impeller, support ring, friction.

Introduction: A number of exploitation methods are used to extract oil from oil fields. CESP is especially used in irrigated areas. Through this pumping unit, a large amount of oil and water in the oil is removed to the surface due to the energy of the mechanically mixed electric motor. One of the main features that distinguishes CESP from other centrifugal pumps is that its parts are multi-section. This pumping unit is a multi-stage, sectional design, and the movement is carried out at the expense of energy transmitted from the electric motor via a cable. The structure of the unit is shown in Fig. 1. This structure includes a ground equipment control station, well equipment and a transformer substation. CESP can be 15-40 m long, depending on the sections connected to it [1].

The underground nodes of the CESP include a three-phase asynchronous electric motor equipped with a tread, a multi-stage centrifugal pump, a check valve and a set of cables, which are shown in Fig 2. These pumping devices can be used in both deep and inclined wells. However, some issues such as failures in the application of these pumping units, accidents, reduction of time between repairs are still relevant today. Currently, 57 oil and gas condensate fields are being exploited in the country, and 20 of these fields are located on the Absheron Peninsula.

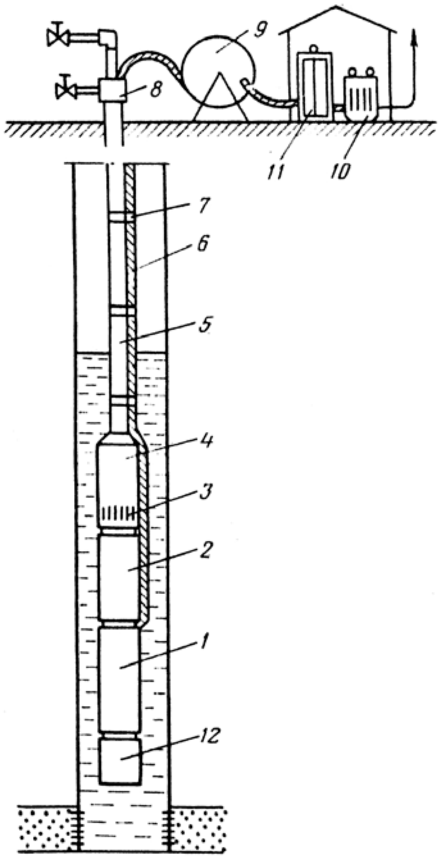


Figure 1. General scheme of centrifugal submersible electric pumping unit
1- electric motor; 2-water protection; 3-pump; 4-cable; 5- control center; 6-way release valve; 7- belt; 8-pump compressor pipes; 9-well surface equipment; 10-transformer; 11-check valve; 12-gas separator.

The application of these pumping units in oil production has a number of advantages [2].:

- Possibility of obtaining high debit in the well during the last operation
- Simplicity of service
- Possibility of application in small diameter of belts used in operation
- Possibility of placement in a small area at the wellhead
- Long maintenance interval compared to rod pumping units.

In addition to the advantages, there are disadvantages, which are as follows:

- Due to the fact that the pumping unit is located at the bottom of the well, it creates difficulties during repair and restoration work.
- The electricity consumed depends on the cost of the product.
- As the operation process takes place at high speed, it causes corrosion in the parts and nodes of the pump under the influence of sand and solid phases.
- A large service center should be established for repair and restoration work.

Through Internet research and analysis of the available literature, a number of results have been obtained that the main causes of accidents and failures are friction and corrosion. These

corrosion causes radial shocks and vibrations during pump operation. At the same time, during the operation, due to the influence of the environment, the insulation layer of the power cable melts and breaks down, filling the guide device and the feathers on the impeller with clay and sand particles [3]. Corroded parts, sand-filled between the blades, a broken impeller and a guide are shown in Fig 3.

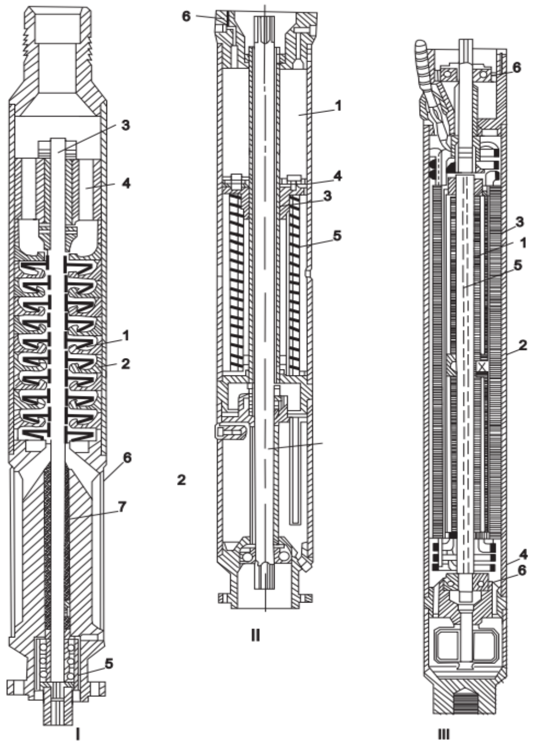


Figure 2. The main parts of CESP, I- Pump; II- Hydro-protection system (tread); III- Engine.



Figure 3. Corroded parts.

Based on the information obtained, we can say that the correct selection of materials is one of the most important issues for CESP. Particular attention should be paid to the materials of the parts in the structures of the main working bodies. In this regard, based on long-term experience in the repair and restoration departments of oil fields, polyamide P-610 plastic mass was used for the impeller of the pump, and high-quality cast iron was used for the guide device. High-tech rubbers were used to obtain the support rings of the impeller, and high-quality stainless steel was used for the pump shaft. In rare cases, texolite material is used in the support rings of the impeller. The construction of CESP, which is applied in the territory of the republic, is a single-tiered working step. A schematic of a single-stage operating pumping unit is shown in Fig. 4 below.

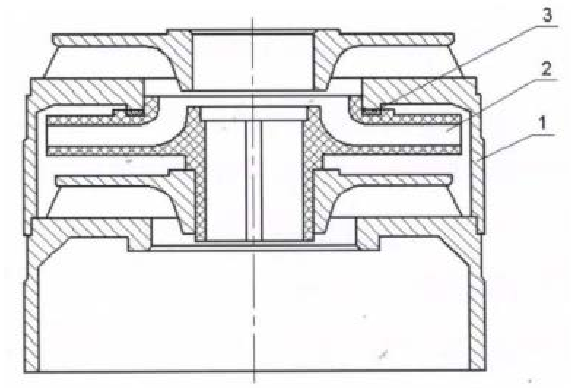


Figure 4. One-piece support ring, 1- guiding part; 2-propeller; 3-support ring.

As shown in Fig 4, a support ring made of texolite material is placed on the surface of the impeller with a guide to prevent friction. Given the current problems, the application of a second support ring can be considered the most effective method to reduce friction and increase the service life of pumps. This structure is shown in Fig. 5a, where 1 and 2 are the support rings. As a result of our idea, the support surface between the impeller and the guide device will be increased, which in turn will increase the service life of the pump. If necessary, it is possible to achieve higher productivity by increasing the number of support surfaces by 2 or 3 times. There are also other advantages of multi-layered work steps, according to research [4].

- can provide reliable protection of the shaft in abrasive and corrosive-aggressive environments.
- As a result of increasing the distance between the working steps, the radial stability of the shaft increases.
- ensures normal operation of the pump in the conditions of large gas factor, unstable dynamic level.

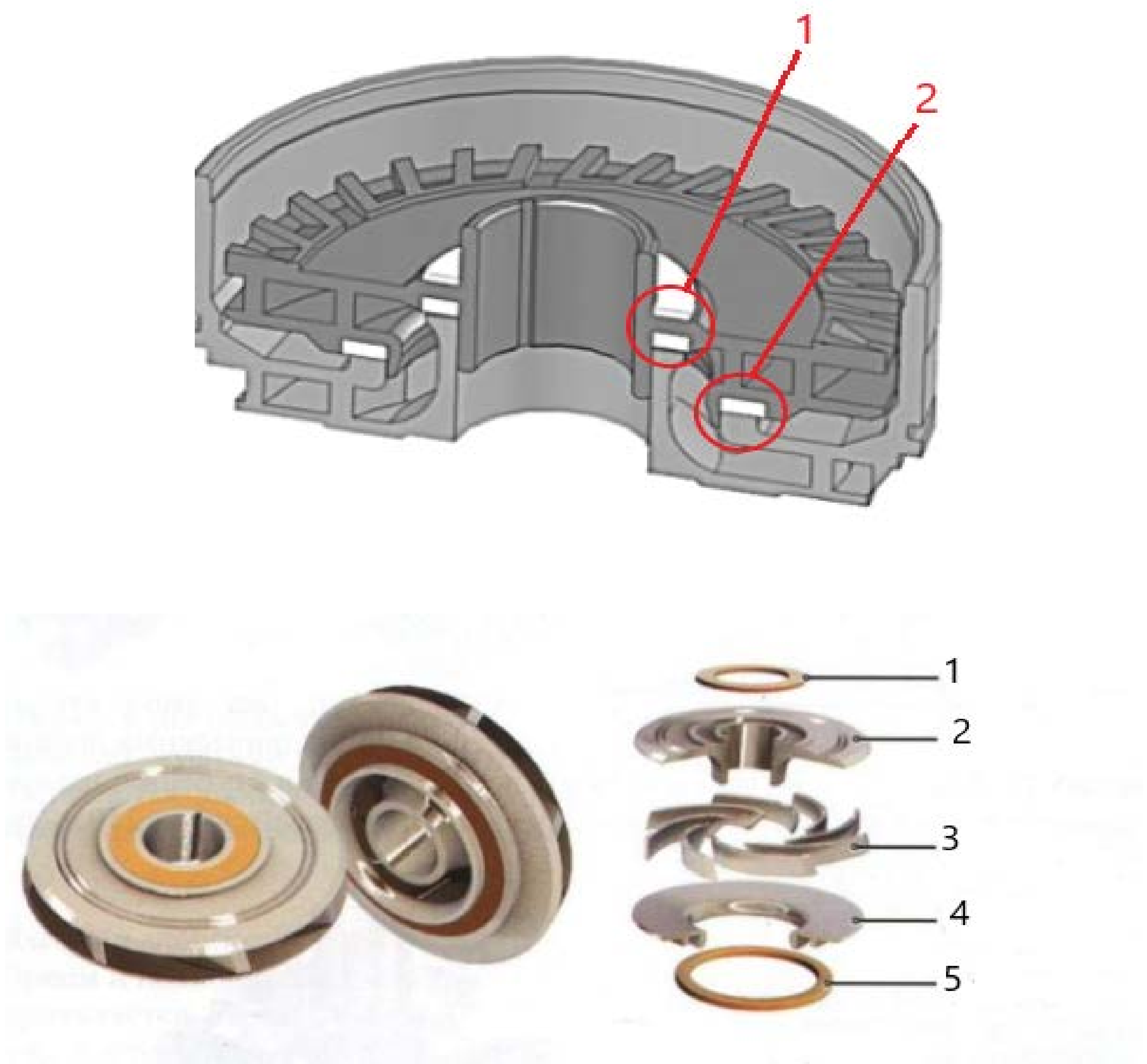


Figure 5a və 5b. Two support rings, 1-upper support ring; 2- upper disk; 3- impeller; 4- lower disk; 5- lower support ring.

By increasing the propeller support ring from one to two, we increase the service life of the pump by an average of 21-58%. This result, along with the improvement of the pump part in CSEP, expands its application.

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MƏRKƏZDƏNQAÇMA DALMA ELEKTRİK NASOS QURĞUSUNUN ƏSAS ELEMENTLƏRİNİN İŞ QABİLİYYƏTİNİN ARTIRILMASININ TƏDQİQİ

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XÜLASƏ

Məqalə istismar prosesi zamanı mərkəzdənqaçma dalma elektrik nasos qurğularında (MDENQ) baş verən imtinaların səbəblərinin araşdırılması, tədqiqi, imtinaların aradan qaldırılması, iş qabiliyyətinin yaxşılaşdırılması və təmirlərarası müddətin artırılması üçün görülən tədbirlərə həsr edilmişdir. MDENQ-nın iş qabiliyyətinin yaxşılaşdırılması məqsədilə konstruksiyada edilən dəyişikliklər öyrənilmişdir.
Açar sözlər: mərkəzdənqaçma dalma elektrik nasos qurğusu, istiqamətləndirici aparat, dayaq halqası, sürtünmə

ИССЛЕДОВАНИЯ ПОВЫШЕНИЯ ПРОИЗВОДИТЕЛЬНОСТИ ОСНОВНЫХ ЭЛЕМЕНТОВ ПОГРУЖНОГО ЦЕНТРОБЕЖНОГО ЭЛЕКТРОНАСОСНОГО УСТРОЙСТВА

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РЕЗЮМЕ

Статья посвящена исследованию причин отказов центробежных погружных электронасосных агрегатов в процессе работы, принимаемым мерам по устранению отказов, повышению производительности и увеличению межремонтного периода. С целью повышения производительности в конструкцию был внесен ряд изменений.
Ключевые слова: центробежный погружной электронасосный агрегат, аппаратура наведения, рабочее колесо, опорное кольцо, трение.

EQUIPMENT. TECHNOLOGIES. MATERIALS

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ALLOCATION OF CAPITAL INVESTMENTS OF AN OIL COMPANY BASED ON INDEPENDENT OPPORTUNITY INFORMATION

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ABSTRACT

To solve the problem of investment allocation with optimization of the total level of return on investment for projects under conditions of risk associated with uncertainty of project income, a linear combination with non-negative coefficients of two loss-making models was used: the model of the usual minimax loss-making and the model of the minimax loss-making level. The problem of probabilistic programming obtained as a result of this combination, provided that the probabilistic variables characterizing the profitability of investments in projects are independent (unrelated), is reduced to a linear programming problem. The solution of this problem by the simplex method gives a vector of the equity distribution of investments that determine the total return of investments by the criterion of the minimum of the loss function.
On the basis of the proposed approach to the optimal solution of the problem of investment allocation, an algorithm for project investment planning has been developed using a database of invested and returned funds for projects in previous planning periods, according to which distributions of possible investment returns are modeled.
The numerical implementation of the algorithm is illustrated by the example of investment planning in an oil company for oil and gas production projects.
Keywords: probability programming, investments, investment portfolio, minimax criterion, loss-making function, return on investment, income, concentrated distribution, probability distribution, measure of opportunity, measure of necessity, fractile model, probability variable, fuzzy variable membership function.

Introduction: When planning investments, the central task is to assess their profitability and risk level. The risk of accepting investment projects is a consequence of unavoidable information uncertainty. The possibility of risk, i.e. a situation in which a project recognized as sound during development will turn out to be unprofitable during implementation, arises in cases when the values of parameters achieved during the investment process deviated from the planned ones or any factors were not taken into account. However, the investor is obliged to try to measure the risks of investment decisions made both at the stage of project development and in the process of its implementation. In the well-known Markovitz model of investment distribution [1], the expected return of an investor is considered as a random variable. Based on this model, the solution of the problem of investment portfolio allocation (portfolio selection) by stochastic programming methods according to the criterion of the minimum variation of the total expected income is obtained under the condition that the maximum average total expected income is preferred..

Probabilistic programming is analogous to stochastic programming. Consequently, its application to the problem of investment allocation gives the expected result, while the expected project revenues are considered as possible variables. The use of probability programming has two main advantages [2]: 1) it is easy to take into account expert information in income estimation; 2) the problem of risk minimization is treated much easier in it than in stochastic programming. However, in most existing approaches using probability programming, independent probability variables are considered, which made it difficult to apply them to solving the problem of investment allocation.

Objective: To construct a non-negative linear combination of the above two models of minimax loss resulting in the distribution of the investment vector $\boldsymbol{c} = (c_1, \dots, c_n)$ with the equity vector $\boldsymbol{x} = (x_1, \dots, x_n)$, in which the total investment $c^T x$ gives the minimum value of the loss function $r(\boldsymbol{x}, \boldsymbol{c})$. It is assumed that $\boldsymbol{e}^T \boldsymbol{x} = 1$, $\boldsymbol{x} \geq 0$, $\boldsymbol{e} = (e_1, \dots, e_n)^T$ and the investment probability variables c_i $(i = 1, \dots, n)$ are independent of each other and have normal distributions with mean m_i and variance σ_i^2. The numerical implementation of the resulting linear combination is demonstrated by the example of the distribution of annual capital investments of an oil company for oil and gas projects.

The results of the study: Consider the distribution of investments of the SOCAR oil company (State Oil Company Azerbaijan Republic) on projects: *i*=1 – AZNEFT (oil); *i*=2 – JV/LTDD (oil); *i*=3 – AZNEFT (gas); *i*=4 – JV/LTDD (gas).
Let $Q_{ij}^{пл}$ and $Q_{ij}^{ф}$ be the planned and actual production volumes in the *i*–th project and *j*-th year (*j*=1,2,...,*m*; *m*=2; *j*=1 - 2017, *j*=2 – 2018, *j*=3 – 2019, *j*=4 – 2020), and the volumes of oil are set at 1000 tons, and gas at 1000 m^3 (see Table 1). In the table.2 oil and gas prices are set: for oil – in \$ /barrel, for gas – in \$ /1000 ^{3} (1 ton of oil ≈ 6.6 barrel).

Table 1. Planned and actual production

	Oil (in thousand tons)			
	2017	2018	2019	2020
AZNEFT	$\frac{6327,6}{6169,6}$	$\frac{6200}{6255,9}$	$\frac{6346,3}{6309,8}$	$\frac{5965,8}{6109,9}$
JV/LTDD	$\frac{1311,5}{1210,9}$	$\frac{1352,1}{1286,4}$	$\frac{1336,8}{1297,3}$	$\frac{1240}{1297,3}$
Газ (в млн. м^3)				
AZNEFT	$\frac{6169,6}{5905,4}$	$\frac{5520}{5662,9}$	$\frac{5619,3}{5912,7}$	$\frac{5913,1}{5922,2}$
JV/LTDD	$\frac{1210,9}{1159,3}$	$\frac{900}{862}$	$\frac{1199,3}{1491,3}$	$\frac{1333,9}{1421,3}$
$Q_{ij}^{пл}$ – in the numerator; $Q_{ij}^{ф}$ – in the denominator				

Table 2. Oil and gas prices

Years	Oil, $/barrel	Gas, $/1000 m^3
2017	52,1	180
2018	73,3	185,2
2019	60,8	285,5
2020	43,7	443,3
2021	71,6	846,3
1$=1,7 manat		

Let $d_j^{(н)}$ be the price of oil in $/barrel in year *j* and $d_j^{(г)}$ be the price of 1000 m^3 of gas in year *j* in *$*. Taking into account the rate of *$* 1=1.7 manat, the price in manats of 1000 tons of oil in the *j* -th year, $Ц_j^{(н)}$, will be written as $Ц_j^{(н)} = 1{,}7 \cdot 6{,}6 \cdot 10^3 d_j^{(н)}{}_i^{(г)}$, and for 1000 m^3 gas $Ц_j^{(г)}$ – in the form of $Ц_j^{(г)} = 1{,}7 \cdot d_j^{(н)}$. Then the invested funds (in manats) in the *j*-th year in projects *i*=1,2 will be written as $S_{ij}^{пл} = Q_{ij}^{пл} \cdot Ц_j^{(н)}$, and for projects *i*=3,4 - $S_{ij}^{пл} = Q_{ij}^{пл} \cdot Ц_j^{(г)}$.
Denote by $c_{ij} = S_{ij}^{ф} - S_{ij}^{пл}$ – the investor's income (in manats) for the project i in year j (with $S_{ij}^{ф} < S_{ij}^{пл}$ means the investor's loss). Taking into account the data in Tables 1 and 2, we get:

$$c_{11} = 3.606513{,}6 \cdot 10^3,\ c_{12} = 5.145014{,}8 \cdot 10^3,\ c_{13} = 4.304394{,}0 \cdot 10^3,$$
$$c_{14} = 2.995769{,}4 \cdot 10^3,\ c_{21} = 707846{,}11 \cdot 10^3,\ c_{22} = 1.057968{,}8 \cdot 10^3,$$
$$c_{23} = 884986{,}9 \cdot 10^3,\ c_{24} = 636084{,}34 \cdot 10^3,\ c_{31} = 1.807052{,}4 \cdot 10^3,$$
$$c_{32} = 1.782624{,}0 \cdot 10^3,\ c_{33} = 2.869728{,}8 \cdot 10^3,\ c_{34} = 4.463029{,}0 \cdot 10^3,$$
$$c_{41} = 354745{,}8 \cdot 10^3,\ c_{42} = 271392{,}08 \cdot 10^3,\ c_{43} = 578051{,}85 \cdot 10^3,$$
$$c_{44} = 1.071105{,}8 \cdot 10^3.$$

From where we find

$$c_{1,\min} = 2.995769{,}4 \cdot 10^3,\ c_{1,\max} = 5.145014{,}8 \cdot 10^3,\ c_1^c = 4.218276{,}6 \cdot 10^3,$$
$$c_{2,\min} = 636084 \cdot 10^3,\ c_{2,\max} = 1.057968{,}8 \cdot 10^3,\ c_2^c = 847026{,}4 \cdot 10^3,$$
$$c_{3,\min} = 1.782624 \cdot 10^3,\ c_{3,\max} = 4.463029 \cdot 10^3,\ c_3^c = 3.122826{,}5 \cdot 10^3,$$
$$c_{4,\min} = 271392{,}08 \cdot 10^3,\ c_{4,\max} = 1.071105{,}8 \cdot 10^3,\ c_4^c = 671248{,}9 \cdot 10^3,$$
$$w_1 = 537311{,}35 \cdot 10^3,\ w_2 = 105471{,}07 \cdot 10^3,\ w_3 = 670101{,}25 \cdot 10^3,$$
$$w_4 = 199928{,}45 \cdot 10^3.$$

For h^0 =0.9, the values r_i^L and r_i^R are calculated using the formulas
При $h^0 = 0{,}9$ величины и рассчитываются по формулам

$$r_i^L = c_i^c - (w_i \ln 10)^{1/2}, \quad r_i^R = c_i^c + (w_i \ln 10)^{1/2}, \quad \ln 10 \approx 2{,}30259.$$

From where we find

$$r_1^L = 4.217924{,}9 \cdot 10^3,\ \ r_1^R = 4.253403{,}6 \cdot 10^3,$$
$$r_2^L = 847010{,}82 \cdot 10^3,\ \ r_2^R = 847038{,}98 \cdot 10^3,$$
$$r_3^L = 312274{,}08 \cdot 10^3,\ \ r_3^R = 3.122838{,}9 \cdot 10^3,$$
$$r_4^L = 671227{,}45 \cdot 10^3,\ \ r_4^R = 671270{,}35 \cdot 10^3.$$

From the solution of system, with the values r_i^L and r_i^R (i = 1, 2, 3, 4) found, we obtain

$$x_1 = 0{,}2703,\ \ x_2 = 0{,}2525,\ \ x_3 = 0{,}2301,\ \ x_4 = 0{,}2471.$$

From table 2 for $m + 1 = 2021$ found $d_{m+1}^{(н)} = 71{,}6$ и $d_{m+1}^{(г)} = 846{,}3$.

Then $Ц_{1,m+1}^{(н)} = Ц_{2,m+1}^{(н)} = 803{,}352$ (manat), $Ц_{3,m+1}^{(г)} = Ц_{4,m+1}^{(г)} = 1438{,}71$ (manat).
And

$$\tilde{S}_{1,m+1} = 5965{,}8 \cdot 803{,}352 = 4.792637{,}3;$$
$$\tilde{S}_{2,m+1} = 1240 \cdot 803{,}352 = 996156{,}48;$$
$$\tilde{S}_{3,m+1} = 5913{,}1 \cdot 1438{,}71 = 8.507236{,}1;$$
$$\tilde{S}_{4,m+1} = 1333 \cdot 1438{,}71 = 1.917800{,}4.$$

Using the following we will determine the total values S_{m+1}

$$S_{m+1} = \sum_{i=1}^{2} \left(Q_{i,m}^{ф} - \tilde{Q}_{i,m}\right) \cdot Ц_{i,m+1}^{(н)} + \sum_{i=3}^{4} \left(Q_{i,m}^{ф} - \tilde{Q}_{i,m}\right) \cdot Ц_{i,m+1}^{(г)}$$

From where we find $S_{m+1} = 300574{,}59$.
Additional investments in the *i*-project in the (*m*+1)-th period $S_{i,m+1}^{доп}$, calculated by the ratio:

$$S_{i,m+1}^{доп} = S_{m+1} \cdot x_i\ (i = 1, 2, 3, 4)$$
$$S_{1,m+1}^{доп} = 81245{,}311,\ \ S_{2,m+1}^{доп} = 75895{,}083,\ \ S_{3,m+1}^{доп} = 69162{,}213,$$
$$S_{4,m+1}^{доп} = 7243{,}879.$$

Then the forecast values of total investments in the (m+1)-th period are equal (in manats)

$$S_{1,m+1}^{tot} = 4.873882{,}6,\ \ S_{2,m+1}^{tot} = 1.072051{,}5,\ \ S_{3,m+1}^{tot} = 8.576398{,}3,$$
$$S_{4,m+1}^{tot} = 1.918524{,}2.$$

Conclusion: 1. The method of modeling the minimax level of unprofitability is one of the new methods of probabilistic programming for solving the problem of investment allocation in conditions of risk associated with uncertainty and insufficient information about the income of investments in projects. It is shown that with the independence of the probability variables describing returns, the problem of the investment portfolio will be reduced to a linear programming problem solved by the well-known simplex method.

2. Since in this work, by defining the loss function $r(x,c)$ for a given share vector $x=(x_1,\dots,x_n)$, $x\geq 0$, $\sum_{i=1}^{n} x_i=1$distribution of the investment vector $c=(c_1,\dots,c_n)$ for projects i =1,...*n*, the largest of the losses is minimized, the resulting solution should be the most preferable, in contrast to the concentrated solution $x_{i_0}=1$, $x_i=0$, $i\neq i_0$, which can obviously be unprofitable.
3. In this paper, a method for choosing a non-negative linear combination is proposed, as a result of which the problem of investment distribution is reduced to a linear programming problem, to which the simplex method is directly applicable.

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РАСПРЕДЕЛЕНИЕ КАПИТАЛЬНЫХ ВЛОЖЕНИЙ НЕФТЯНОЙ КОМПАНИИ НА ОСНОВЕ НЕЗАВИСИМОЙ ВОЗМОЖНОСТНОЙ ИНФОРМАЦИИ

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РЕЗЮМЕ

Для решения задачи распределения инвестиций с оптимизацией общего уровня доходности в условиях риска, связанного с неопределенностью, использовалась линейная комбинация с неотрицательными коэффициентами двух убыточных моделей: модели обычной минимаксной убыточности и модели минимаксный уровень убыточности. Задача вероятностного программирования, полученная в результате такой комбинации, при условии, что вероятностные переменные, характеризующие доходность инвестиций в проекты, являются независимыми (не связанными), сводится к задаче линейного программирования. Решение этой задачи симплексным методом

дает вектор распределения капитала инвестиций, определяющий общую доходность инвестиций по критерию минимума функции потерь.
На основе предложенного подхода к оптимальному решению задачи распределения инвестиций разработан алгоритм планирования инвестиций в проекты с использованием базы данных вложенных и возвращенных средств по проектам в предыдущие периоды планирования, в соответствии с которым моделируются распределения возможной доходности инвестиций.
Численная реализация алгоритма проиллюстрирована на примере инвестиционного планирования в нефтяной компании для проектов по добыче нефти и газа.
Ключевые слова: вероятностное программирование, инвестиции, инвестиционный портфель, минимаксный критерий, функция убыточности, рентабельность инвестиций, доход, концентрированное распределение, распределение вероятностей, мера возможностей, мера необходимости, фрактильная модель, вероятностная переменная, функция принадлежности нечеткой переменной.

EQUIPMENT. TECHNOLOGIES. MATERIALS

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ABOUT THE METHOD OF CALCULATION THE PUMJACK`S BALANCING

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ABSTRACT

The consistent performance of a modern oil-well pumping unit depends on many factors. One of them is equalizing the operation of the electric motor. The article considers the equilizing method applicable for double lever pumping units manufactured according to the industry standard of Azerbaijan OST 26-16-08-87. The study is based on the idea that the work done on the crankshaft of the pumping unit per cycle is the sum of the work done during the upward stroke and the work performed during the downward stroke of the rod suspension point (RSP). Formulas for determining the weight of balancing masses are presented.
Key words: driver, pumping unit, electric motor, equilizing, equilizer, crank, torque.

The relevance of research: Despite the fact that for about 100 years, as a rule, balancing pumping jacks have been used as a drive for oil production by pumping, the need to improve the drives of borehole rod pumps (BRP) does not lose its relevance at the present time.
The goal of improvement is chosen depending on the needs of society. For example, the transition from the axial kinematic scheme of two-arm balancing pumping jacks that existed until 1986 to a de-axial scheme made it possible to save metal consumption for the manufacture of a pumping unit by reducing the overall dimensions, as well as to reduce the power of the electric motor. This article discusses the method of balancing the work of the electric motor.

Introduction: Among balancing pumping jacks produced by different countries and companies, there are one-arm, two-arm, axial and de-axial kinematic schemes. One of the important tasks in the design of BRP drives is balancing the operation of the electric motor, on the uninterrupted operation of which the performance of the entire installation depends.
For balancing pumping jacks, balancing the operation of the electric motor is achieved by installing counterweights either on the balance beam or on the crank, and the combined balancing method is also used. The required weight of the load, as well as the installation location, depends on the selected mode of operation of the entire downhole rod pumping unit (DRPU).

Main part: Balancing the work of mass-produced pumping units can be approached in accordance with the following principles:
1) achieving equality of work performed by the electric motor of the pumping unit when moving up and down the suspension point of the rods (RSP);
2) achieving equality in highest torque for both half-cycles;
3) achieving the lowest RMS torque per cycle.

The third principle is the most correct in terms of improving the working conditions of the electric motor, the second principle is in terms of improving the working conditions of the pumping unit gearbox. The first principle cannot be strictly substantiated and was used only to derive the simplest formulas for calculating the weight of counterweights. This principle has not received significant application. The third principle has received the greatest development. Some recognition was also given to the second principle.
As a general case, a combined (balance and crank) balancing method is adopted to derive a formula for balancing the operation of a pumping unit.
The leading link of the transforming mechanism of the pumping unit is the crank. A torque $M_{кр}$ is applied to the crank through the driven shaft of the gearbox, the value of which is determined by the values of all types of resistance that must be overcome, and is equal to:
$M_{кр} = T \cdot r$

where $M_{кр}$ is the tangential component of the force acting on the crank and changing depending on the angle of rotation of the crank φ; r - is the radius of the crank, i.e. distance from the center of rotation of the crank to the point of connection with the connecting rod.
The work done on the crankshaft of the pumping unit per cycle is the sum of the work done on the upstroke and the work done on the downstroke of the RSP. Methods for calculating the reduced weight of balancing masses are based on the principle of equality of work for half-cycles. Equating these works, one can obtain an equation from which the value of the required reduced weight of counterweights $T_{КП}$ is determined:

$$\int_{\varphi_{кн}}^{\varphi_{кв}} T_{КП} r d\varphi_1 = \int_{\varphi_{кв}}^{2\pi+\varphi_{кн}} T_{КП} r d\varphi_1 \qquad (1)$$

where $\varphi_{кн}$ and $\varphi_{кв}$ are the angles of rotation of the crank corresponding to the lower and upper extreme positions of the RSP.
In the considered method, this conclusion is given for a particular case, when $\varphi_{кв} = 0$, and $\varphi_{кн} = \pi$, i.e. for a pumping unit with a symmetric (axial) kinematic scheme (de-axial angle θ = 0o) [1].
Modern mass-produced pumping units according to OST 26-16-08-87 are manufactured according to an asymmetric (de-axial) kinematic scheme with a de-axial angle θ = 9 degrees. Since the displacement of the center of rotation of the crank from the vertical connecting the extreme upper and lower points of connection of the connecting rod with the balancer is insignificant and will not significantly affect the value of the balancing load, it was decided in the first approximation not to take into account the value of this displacement. Also considering that according to the existing standard, a type of pumping unit with a combined balancing system (СКД3-1,5-710) is provided, the conclusions are given to take into account balancing and crank balancing.
The essence of balancing the work of the pumping unit, necessary for the normal unloaded operation of the electric motor, is as follows. When moving down the suspension point of the rods (RSP), the electric motor of the pumping unit expends work on lifting the counterweight and the string of sucker rods AN, and when the RSP moves upwards, the counterweight

returns this work, helping the electric motor to lift the rod string and liquid AB. Thus, one can write:

$$A_B = \int_0^{\pi} T_{\text{КП}}(\varphi) r d\varphi_1 = r \int_0^{\pi} T_{\text{КП}}(\varphi) d\varphi_1$$

$$A_H = \int_{\pi}^{2\pi} T_{\text{КП}}(\varphi) r d\varphi_1 = r \int_{\pi}^{2\pi} T_{\text{КП}}(\varphi) d\varphi_1$$

To determine the value of the balancing load, the condition is accepted that the work performed by the pumping unit, and, consequently, the load on the electric motor during the upward stroke will be equal to the load during the downward stroke. Equating, we get:

$$\int_0^{\pi} T_{\text{КП}}(\varphi) r d\varphi_1 = \int_{\pi}^{2\pi} T_{\text{КП}}(\varphi) d\varphi_1$$

On the kinematic diagram of the pumping unit with the necessary designations of angles, links of the transforming mechanism and forces is presented. Based on the principle of possible displacements, we can write the following expression for the tangential force on the crank with combined balancing (Fig. 1)

$$T_{\text{КП}} = \frac{k_1}{\text{k}} - P_{\text{тпш}} \cdot \frac{V_B}{\omega r} - \frac{L_{\text{Б}}}{k} F_{\text{Б}} A \frac{V_B}{\omega r} - F_{\text{К}} \sin\varphi \qquad (2)$$

where k_1 is the length of the front arm of the balance bar; k is the length of the rear arm of the balancer; $P_{\text{тпш}}$- force at the suspension point of the rods (RSP); $F_{\text{Б}}$ is the weight of the balancing counterweight; $L_{\text{Б}}$- arm of the balancing counterweight; V_B - speed of point B of the composition of the balancer with the connecting rod; W is the angular speed of rotation of the crank; r is the length of the crank radius; $F_{\text{К}}$- the weight of the crank balancing masses reduced to the point of articulation of the connecting rod with the crank; φ is the angle of rotation of the crank from the upper vertical position; A is a coefficient that takes into account changes in the action of the balancing counterweight during the cycle of operation of the mechanism due to the influence of the forces of inertia of this counterweight and due to the movement of the latter not along the vertical, but along the arc:

$$A = \cos\delta_1 - \frac{L_{\text{Б}}}{k} \frac{\text{a}_B}{g} \qquad (3)$$

where δ_1 is the angle of deviation of the balance bar from the horizontal; a_B is the acceleration of point B; g is the acceleration due to gravity.

The second expression (2) is valid for pumping units with swinging masses statically balanced relative to the balancer axis (not counting the balancing counterweight) and does not take into account the effect of the swing masses of the balancer itself on the tangential force on the crank. For modern pumping units with the selected pumping mode, both of these assumptions do not give any significant errors in the determination of the tangential force on the crank.

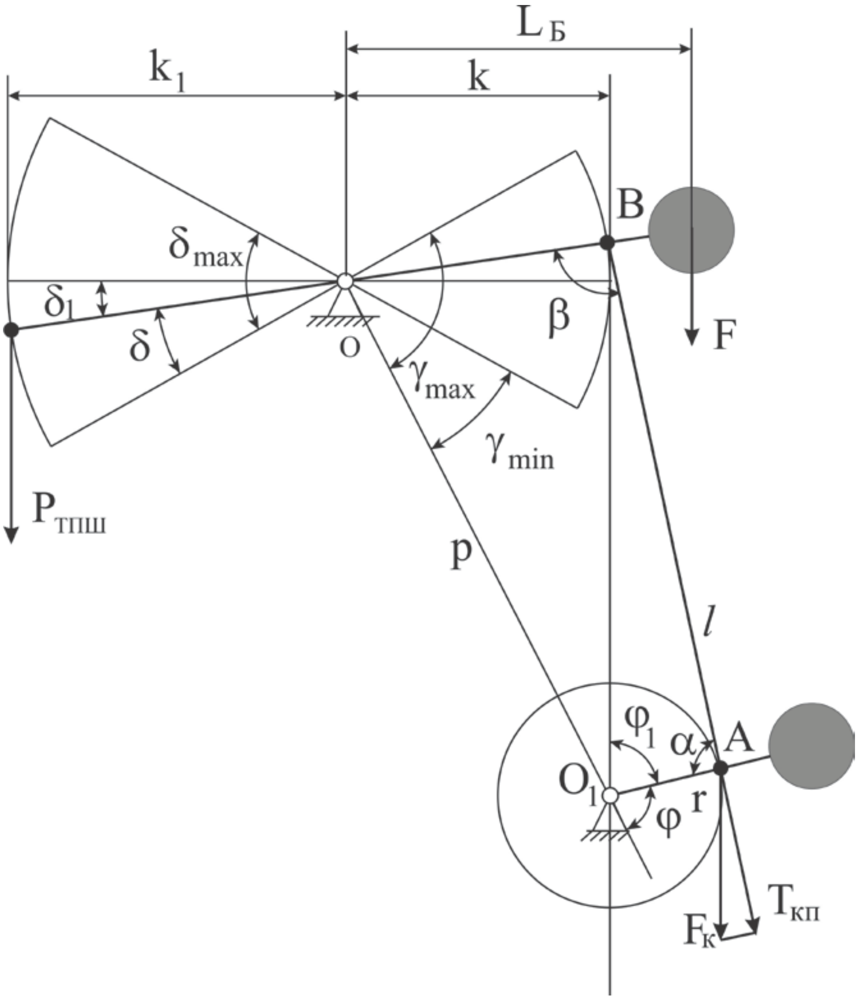


Figure 1. Scheme for calculating the balance of pumping unit.

Some static disbalance of the oscillating masses of the pumping unit itself can be taken into an account by a corresponding increase or decrease in the weight of the balance weight substituted into formula (2).

After substituting the value T into equation (3) according to formula (1), substituting the value A according to formula (2), we will have:

$$L_{Б}F_{Б}\left(\int_{\varphi_{кн}}^{\varphi_{кв}}\cos\delta_1\frac{V_B}{\omega k}d\varphi-\int_{\varphi_{кв}}^{2\pi+\varphi_{кн}}\cos\delta_1\frac{V_B}{\omega k}d\varphi-\frac{L_{Б}}{k}\int_{\varphi_{кн}}^{\varphi_{кв}}\frac{a_B}{g}\frac{V_B}{\omega k}d\varphi+\frac{L_{Б}}{k}\int_{\varphi_{кв}}^{2\pi+\varphi_{кн}}\frac{a_B}{g}\frac{V_B}{\omega k}d\varphi)\right)$$

$$+$$

$$+rF_{КП}\left(\int_{\varphi_{кн}}^{\varphi_{кв}}\sin\varphi d\varphi-\int_{\varphi_{кв}}^{2\pi+\varphi_{кн}}\sin\varphi d\varphi\right)=k_1\left(\int_{\varphi_{кн}}^{\varphi_{кв}}P_{ТПШ}\frac{V_B}{\omega k}d\varphi-\int_{\varphi_{кв}}^{2\pi+\varphi_{кн}}P_{ТПШ}\frac{V_B}{\omega k}d\varphi\right)\quad(4)$$

From fig.1 we get: $\delta_1=\frac{\delta_{max}}{2}-\delta$

Which gives us $\frac{d\delta_1}{dt}=-\frac{d\delta}{dt}=-\frac{V_B}{k}$

Substituting instead of $\frac{V_B}{k}$ in equation (4) the value of this ratio according to equation (5), taking into account that dφ/dt=ω and replacing the integration limits, we will have:

$$L_БF_Б\left(\int_{\delta_{max/2}}^{-\delta_{max/2}}\cos\delta_1 d\delta_1-\int_{-\delta_{max/2}}^{\delta_{max/2}}\cos\delta_1 d\delta_1+\frac{L_Б}{k}\int_0^{\delta_{max}}\frac{a_B}{g}d\delta-\frac{L_Б}{k}\int_{\delta_{max}}^{0}\frac{a_B}{g}d\delta\right)+2rF_{КП}(\cos\varphi_{кн}-\cos\varphi_{кв})=k_1\left(\int_0^{\delta_{max}}P_{ТПШ}d\delta-\int_{\delta_{max}}^{0}P_{ТПШ}d\delta\right)\quad(6)$$

For pumping units with crank balancing, due to the absence of balancing in formula (6), the first polynomial will be equal to zero. Then for crank balancing we get:
To determine the net weight of counterweights, formulas are proposed for the following balancing methods:
- for combined balancing

$$Q_{ку}=\frac{L_Б}{k_1}F_Б+\frac{k}{k_1}\frac{R}{r}F_{КП}$$

- for rotary balancing

$$Q_K=\left(P_ш+\frac{P_ж}{2}\right)\frac{k_1}{k}\frac{r}{R}$$

where R is the radius of installation of the counterweight on the crank.
In pumping units according to OST 26-16-08-87, there are recommendations which are provided for balancing the operation of mass-produced pumping units, based on nomograms, with the help of which the required number of counterweights and the location on the crank are selected depending on the balancing moment of the $M_{УР}$ [2]:

$$M_{ур}=S(P_{шт}+0{,}5P_ж)=S(P_{max}+P_{min})/2$$

where S is the stroke length of the RSP, m; $P_{шт}$ – rod weight in liquid, kg; $P_ж$ is the weight of the liquid column in the tubing above the plunger of the borehole pump, kg; P_{max} and P_{min} - maximum and minimum load in RSP, кгс.

Figure 2 shows a nomogram for choosing the number of counterweights and their installation location on the crank of the СКД6-2,5-2600 pumping unit (the mass of one counterweight is 485 kg).

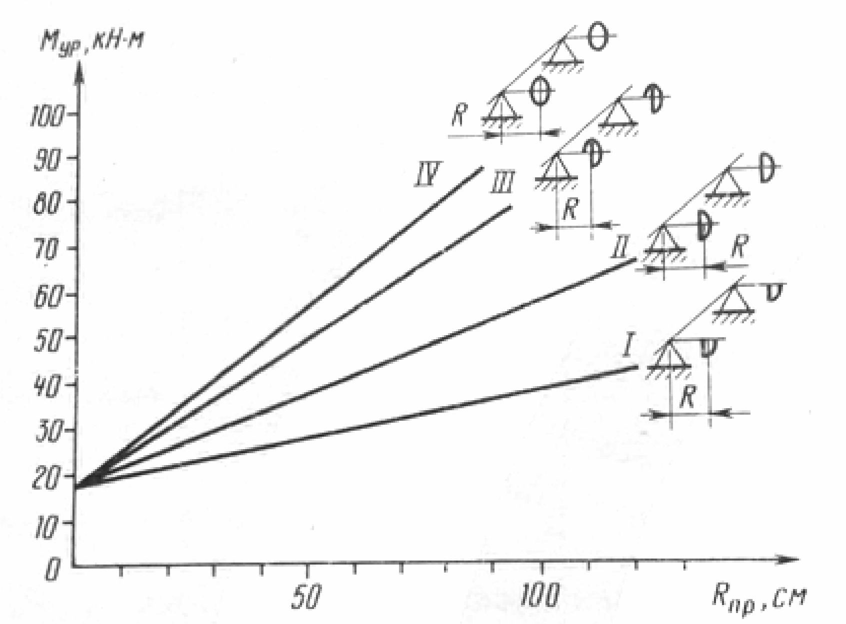


Figure 2. Nomogram for choosing the number of counterweights and their installation location on the crank of the pumping unit SKD6-2.5-2600:
I - one counterweight on each crank; II - two counterweights on each crank; III - three counterweights on each crank; IV - four counterweights on each crank

But nomograms, like formulas, cannot always provide the best balancing of the pumping machine. This is due to the fact that it is theoretically impossible to take into account all the loads that occur in the links of the pumping unit, as well as the degree of deterioration of its nodes, the efficiency of the entire installation, which, as a rule, are not taken into account in theoretical formulas, but significantly affect the balancing of the pumping unit.
Therefore, the final balancing is carried out by controlling the current consumed by the electric motor, when the head of the balancer moves up and down. The arrow of the amperemeter, included in the power line supplying the engine, should give the same maximum deviations when moving up and down.
Nevertheless, when creating new designs of pumping units, it is recommended to use formulas for a preliminary assessment of the possibility of balancing the pumping unit.

Conclusion: The described method for calculating the balancing of the operation of an electric motor is presented for a de-axial kinematic scheme of a real two-arm pumping unit, it allows, for a given load in the RSP, to determine the forces acting at the installation point of the counterweights, and formulas are also presented for determining the weight of the balancing masses for the combined and crank balancing methods of mass-produced pumping units.

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О МЕТОДИКЕ РАСЧЕТА УРАВНОВЕШИВАНИЯ РАБОТЫ СТАНКА-КАЧАЛКИ

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РЕЗЮМЕ

Надежная работа современной скважинной насосной установки зависит от многих факторов, одним из которых является уравновешивание работы электродвигателя. В статье рассмотрен метод уравновешивания, применимый к двуплечим балансирным станкам-качалкам, изготавливаемым по отраслевому стандарту Азербайджана ОСТ 26-16-08-87. За основу исследования взято положение, что работа, произведенная на кривошипном валу станка-качалки за цикл, слагается из работы, произведенной при ходе вверх и работы, произведенной при ходе вниз точки подвеса штанг (ТПШ). Представлены формулы для определения веса уравновешивающих масс.
Ключевые слова: привод, станок-качалка, электродвигатель, уравновешивание, балансир, кривошип, крутящий момент.

MANCANAQ DƏZGAHININ TARAZLAŞMANIN HESABLAMASININ METODİKASI HAQQINDA

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XÜLASƏ

Müasir quyu nasos qurğusunun etibarlı işləməsi bir çox amillərdən asılıdır, bunlardan biri elektrik mühərrikinin işini balanslaşdırmaqdır. Məqalədə Azərbaycanın OST 26-16-08-87

sənaye standartına uyğun olaraq istehsal edilmiş iki qollu sallanan maşınlara tətbiq edilən balanslaşdırma üsulu nəzərdən keçirilir. Tədqiqat belə bir mövqeyə əsaslanır ki, hər dövrədə nasos qurğusunun dirsək valında görülən iş, çubuqun asma nöqtəsinin (TSS) yuxarı vuruşu zamanı görülən işlərin və aşağıya doğru vuruş zamanı görülən işlərin cəmidir. Balanslaşdıran kütlələrin çəkisini təyin etmək üçün düsturlar təqdim olunur.
Açar sözlər: mühərrik, nasos qurğusu, elektrik mühərriki, balanslaşdırma, balanslaşdırıcı, krank, fırlanma momenti.

EQUIPMENT. TECHNOLOGIES. MATERIALS

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CALCULATION OF GEOMETRIC PARAMETERS THE LATERAL WELLBORE

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ABSTRACT

Increasing volumes of drilling of lateral branches of trunks, using the main trunks, also put forward the need to solve a number of new problems. Among them, there are small radial gaps between the production string or liner and the wall of the lateral wellbore, which make it difficult to ensure the tightness of the annulus. The problem is exacerbated by the fact that on an inclined or horizontal section of the wellbore, the production string is always eccentrically displaced relative to the axis of the well. Therefore, ensuring the required quality of sealing may require continuous or local expansion of the trunk.
Keywords: Borehole, drilling, ensuring tightness, descent and lifting operations, chisels, side barrel.

Small radial gaps between the production column or shank and the wall of the lateral borehole make it difficult to ensure the required quality of sealing of the backwater space.
The increasing volumes of drilling of lateral branches of trunks, using the main trunks, also put forward the need to solve a number of new tasks. Among them, there are small radial gaps between the production column or the shank and the wall of the side borehole, which make it difficult to ensure the tightness of the backwater space. The problem is compounded by the fact that on an inclined or horizontal section of the trunk, the production column is always eccentrically displaced relative to the axis of the well. Therefore, ensuring the necessary sealing quality may require continuous (along the entire length of the trunk branch) or local (in separate sections) expansion of the trunk, limited:
- the maximum permissible value, if exceeded, the cement mortar in certain sections of the well will not be able to displace the washing liquid from the narrowest gap between the bottom wall of the trunk and the casing;
- the minimum value, limited by the maximum permissible, for one reason or another, hydraulic resistance of the column space.
Based on the conducted research and analysis of known solutions, formulas for determining the rational intervals of expansion of the lateral borehole are presented. For example, 120 are used to wire a side branch from a 215.9 mm well previously lined with 146 mm casing pipes...124-mm chisels. In the fields, in order to reduce hydraulic losses in the backwater space, it may be necessary to expand the trunk, for example, using the RS-116/137 expander of the AzINMASH design with self-moving cutting elements only when expanding or working out the well (during descent and lifting operations, the diameter of the expander is 116 mm). The maximum permissible diameter of expansion, depending on the rheological

properties of cement and clay solutions, based on the conditions for ensuring the tightness of the column space, should not exceed 130 ... 155 mm.

Based on the nomenclature of the equipment mastered by the industry for completion, exploration and development of wells, the use of production columns from 102-mm casing pipes developed by VNIITNEFT should be considered rational for fixing the lateral branches of the trunks. However, in wells previously lined with 146-mm production columns (most of the existing stock), when using 102-mm pipes, there is a need for additional expansion of the lateral branch of the trunk from 120 ...124 mm to a value that ensures the prevention of leakage of the backwater space observed in the fields. Such expansion can be carried out with the help of sliding spherical or spot expanders.

The choice of the expansion scheme (along the entire length of the trunk, local - with the creation of artificial caverns or a combination of both schemes) is made depending on the conditions of the productive horizon, the patency and equipment of the casing, the presence of possible complications during the completion and subsequent operation of the well and a number of other factors.

Since, regardless of the expansion scheme on the inclined and horizontal sections of the well, the column always "lies" on the bottom wall of the trunk, it is mandatory for all schemes to check the minimum and maximum allowable diameters of the trunk expansion, providing high-quality insulation of the cavern space, taking into account the main geological and technological factors and equipment used for cementing the well.

It is obvious that the maximum expansion diameter is selected based on the condition that the backwater space is completely filled with grouting solution in certain sections of the well (to ensure isolation, first of all, of productive and water protection complexes), taking into account the available equipment of the casing and the well grouting mode. The minimum diameter is based on the permissible maximum hydraulic resistance of the column space when cementing the production column. Let's consider both issues separately.

With the eccentric placement of the site in question, for example, in the zone of a productive complex, an operational column on an inclined or horizontal part of the trunk, the complete replacement of the washing liquid with a grouting solution for viscoplastic liquids (to which most of the cement and clay solutions belong) is provided at some minimum rate of fluid injection into the well [1].

And indeed, in order to exclude the occurrence of stagnant zones during the well plugging process, it is necessary to ensure the movement of liquid in the narrowest part of the eccentric annular space by exceeding, by the influence of the pressure drop P per unit length of the column, the value of the dynamic shear stress of the grouting or clay solution. In this case, the flow of liquid through an elementary segment with an infinitesimal arc length dl = R dω is equivalent to

$$\Delta Q = \frac{Q_k * dl}{2\pi R} = \frac{Q_k * (\varpi)}{2\pi} d\omega \tag{1}$$

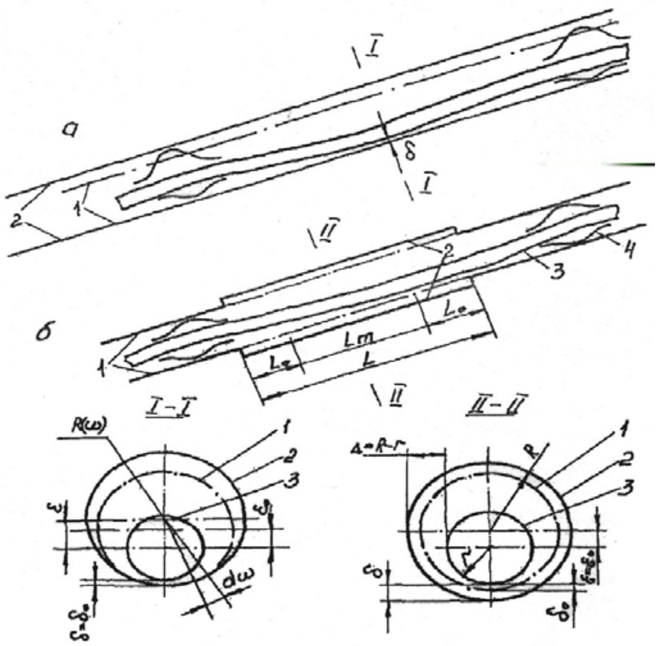


Figure 1. Scheme for the selection of geometric parameters of the lateral borehole expansion: 1 - the initial wellbore; 2 - the expanded trunk: a - the fully expanded trunk; b - the site of local expansion; 3 - maintenance column; 4 - centering lamp.

Substituting the flow rate Qk* [2] through an elementary segment into the resulting expression, integrating it along the entire perimeter of the annular space and discarding negligibly small terms after transformations, we obtain a formula for determining the minimum productivity of cementing aggregates that ensures the flow of grouting mortar through the entire section of the eccentric annular space in the form [1]:

$$Q_k ** \cong \frac{\pi\tau_0}{8\eta}\left[\frac{-\delta^2 r}{6} + \left(\frac{10r}{6} - \frac{\delta}{6}\right)\delta\Delta - \left(\frac{8r}{3} - \frac{3\delta}{4}\right)\left(\Delta^2 + \frac{\varepsilon^2}{2}\right) + \right.$$

$$\left. + \left(\frac{4r}{3\delta} - 1\right)\Delta\left(\Delta^2 + \frac{3\varepsilon^2}{2}\right) + \frac{\Delta^2}{3\delta}\left(\Delta^2 + 3\varepsilon^2\right) - \frac{13\Delta^3}{72r\delta}\left(\Delta^2 + 5\varepsilon^2 + \frac{15\varepsilon^4}{8\Delta^2}\right)\right] \quad (2.)$$

Here, t_0 and η are, respectively, the dynamic shear stress and viscosity of the grouting solution or washing liquid; ε is the permissible or actual (determined depending on the centering equipment of the column in the zone of the productive complex) eccentricity of the casing displacement in the well section under consideration; R is the radius of the well; r is the outer radius of the casing pipes;

5 = R-r-ε; Δ = R-r.

The obtained solution in the special case with the eccentricity of the displacement ε = 0 completely coincides with the original one [2] for determining the flow rate Q_k* under the structural flow regime of viscoplastic liquids in a concentric annular space.

Fig.1. (upper part) shows the effect of the well diameter found by the formula (2.) on the required minimum fluid flow rate, which ensures the flow of some types of cement and clay solutions along the entire perimeter of the section of the eccentric annular space. The resulting graphical dependence makes it easy to determine the diameter of the borehole expansion for a column of 102 mm casing pipes, provided that the eccentricity of the displacement of the column relative to the axis of the well according to the regulatory materials of VNIIKRNEFT (developed at the time for vertical wells) does not exceed half the difference in the diameters of the well and the outer pipes. From Fig. 1 it is clearly seen that the quality of the insulation of the column space, first of all, is affected by the ratio τ_o/η of cement mortar.
The analysis of the obtained solution clearly confirms the inadmissibility of using, in contrast to common regulatory materials, a fixed value of the displacement eccentricity in various sections of the shafts of inclined and horizontal wells. The dominant influence of the ratio τ_o/η shows that the chemical treatment of cement mortars, for example, with sulfite-alcohol bard and its derivatives, has a significant, up to multiple variations, value on the completeness of the replacement of the washing liquid with a grouting solution.

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РАСЧЕТ ГЕОМЕТРИЧЕСКИХ ПАРАМЕТРОВ БОКОВОГО СТВОЛА СКВАЖИНЫ

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РЕЗЮМЕ

Возрастающие объемы бурения боковых ответвлений стволов, с использованием основных стволов, выдвигают и необходимость решения ряда новых задач. Среди них, и малые радиальные зазоры между эксплуатационной колонной или хвостовиком и стенкой бокового ствола скважины, которые затрудняют обеспечение герметичности за колонного пространства. Проблема усугубляется тем, что на наклонном или горизонтальном участке ствола эксплуатационная колонна всегда находится эксцентрично смещенной, относительно оси скважины. Поэтому обеспечение необходимого качества герметизации может потребовать сплошного или локального расширения ствола.
Ключевые слова: Скважина, бурения, обеспечение герметичности, спуско-подъемные операции, долота, боковой ствол.

QUYUNUN YAN LÜLƏSİNİN HƏNDƏSİ PARAMETRLƏRİNİN HESABLANMASI

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XÜLASƏ

Əsas lülədən istifadə etməklə, quyuların bərpa işlərində yan lülənin qazılması texnologiyasının genişlənməsi bir sıra vacib məsələlərin həllini tələb edir.
Bu sırada istismar sütunu və quyunun yan lüləsinin divarı arasında kiçik radial boşluqlar mövcuddur ki, bu da boruarxası fəzanın hermetikliyinin təmin edilməsini çətinləşdirir. Problem quyunun gövdəsinin maili və ya horizontal sahəsində istismar sütununun həmişə ekssentrik olaraq quyunun oxuna nisbətən dəyişilməsi ilə daha da dərinləşir.
Açar sözlər: quyu, qazma, hermetikliyin təmini, enmə-qaldırma əməliyyatları, quyu balta, yan lülə.

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ANALYSIS OF THE SPILL PROCESS OF RECTANGLE SQUARE PASTES USED IN ARMATURE ROLL

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ABSTRACT

The study shows the indirect ways of distorting the profiles of rectangular square passages used in reinforcement or changing the size of individual areas of their perimeter. It was found that if the profile of a pasteurizer is messed up, even if there are no cracks, it can cause a number of problems in the next steps.

Keywords: armature, rolling, pistachio, square, crack, rectangle, defect, pouring process.

The issue's importance: Founded in 2001 by National Leader Heydar Aliyev, Baku Steel Company (BSC) has been maintaining its competitiveness in the world market by constantly increasing its production based on innovation and quality. The company is always adding to what it does. Right now, it has two electric arc furnaces and three rolling lines, which means it can take 1,000,000 tons of steel per year. In addition to increasing production capacity, the BSC, which has been completely reconstructed since 2010, has expanded its production line to produce fittings, square pistols, circular pistols, channels, ropes, angles, doublets, squares, and iron rods, as well as a variety of figured castings.

By completely automating the manufacturing process, the company has removed the chance of accidents caused by people, sped up production, and made sure that the quality of the products is the best it can be. Additionally, as a direct consequence of the high priority given to the environment by the company's management, the Gas-Dust Trapping Device is updated, and the dust and dirty air generated during the production process is cleaned in a fully automated manner. Because of this, there will be no contamination of the environment.

The goal of the study: The method of pouring rectangular square pistachios used in armature rolling is analyzed, and ways to boost productivity are thought up.

An analysis of the current state of production at BSC: The main production begins with the preparation of the charge and its design in accordance with the technological operations in the 60-ton Electric Arc Furnace. The scrap metal entering the facility is sorted. cut with press scissors, sent to the press for briquetting, and from there it is directed to the charge areas in accordance with the standard.

According to the findings of our investigation, 87–9 percent of the final product is derived from various types of metal waste, the majority of which is composed of steel waste (65–7 percent), cast iron (10–7 percent), and HBI or Okatish (DRI) (25–7 percent). The processing of 66–67 tons of scrap metal results in the production of 58–60 tons of ready-made pasta on average.

Since both electric arc furnaces (EQS) use scrap metal, the scrap metal must meet the standard in order to get to the Shikhta region, where it must be put into the baths in a specific way before being taken to the furnace. Baku Steel Company is the most important player in the country's ferrous metallurgy industry. To keep its dominant market position, the company is putting a lot of effort into developing cutting-edge technologies and making scientific and technological breakthroughs. In order to accomplish this goal and boost the level of output produced by the Shikhta field, a cutting-edge scrap press and packaging equipment manufactured by the Italian firm Vezzani were installed. The smelting process, which was manufactured by the German company Siemens VAI, runs in automatic mode so that production can be increased. Additionally, all of the protection systems are fully electronic, and a more modern 60-ton electric arc furnace is used. Additionally, in order to protect the environment and keep the air clean, Demora's automatic gas is used to clean the air that has been polluted. A gadget for removing dust has been put in place.
New technologies for controlling the process of infusing carbonaceous materials into the 60-ton Electric Arc Furnace have been installed. Furthermore, the combined 3 x 4 MWT RCB burners optimize the process of blowing oxygen and carbon into the electric arc furnace, as well as precisely complement the energy provided by gas, oxygen, and carbon exothermic processes. The functioning of the primary power units in the steelmaking process is automatically controlled by software.
Steel shafts get liquid metal from the furnace, and ferroalloys, lime, and other additives are added to the liquid metal throughout manufacture. Components are given to fulfill the steel standard's criteria. The temperature, chemical composition, and other parameters of the liquid metal entering the furnace are all customized. Before putting the liquid metal into an automated continuous paste casting machine, CVS puts it through a rebuilt kiln.
A 6 or 12 meters long (100x100mm to 150x150mm) square paste is poured, cut by automatic cutting, and treated by refrigerated rollers on a continuous casting machine (FPTM) with four nozzles for 35–40 minutes. The pasta is inspected by technical control in the field of pasta processing; flaws in the pasta are disclosed, and the pasta is sent to the reheating furnace of the rolling area and rolling shop in accordance with the order.
The square plates are heated in a reheating furnace and thermomechanically processed into various sizes, such as 10-32 mm fittings, 5.5 mm, 7 mm, and 8 mm diameter rods, depending on the order at the rolling shop. There is a 40-125 mm corner, an 80-140 mm channel, and a 100-140 mm doublet on the rolling line No. 2, which is being rebuilt by the Turkish company ATAMET.
The manufacturing of round pistachios began in January 2016 as part of the 2nd Electric Steel Melting Workshop (EPES-2) reconstruction, which began in 2014. VNIIMETMASH's two-way circular casting machine is outfitted with cutting-edge technology. To avoid contact with the air and pollutants from the environment, circular pistachios are poured into a closed system built of specific refractory materials.
Pistachios with diameters of 130 mm, 160 mm, 170 mm, 190 mm, and 220 mm are currently produced in lengths ranging from 3000 mm to 12000 mm, depending on demand. Because these tools are intended for pipe rolling, high-grade steel grades such as St20, St45.09G2C, 32G2, and others are commonly used.
The company's management, which has always placed a high value on innovation, monitors the quality of our goods from the beginning to the completion of the smelting process.

Spectral analysis instruments are used to monitor pollution. Compressed air transports samples from the furnace, kiln, and continuous casting machine to the laboratory, where the results of the analysis are fed into the control computer.

A variety of things that BSC makes. The company that makes things makes the following things:
Armature. AZS 538-2011 (GOST 52544-2006), GOST P34028 with diameter of 10, 12, 14, 16, 18, 20, 22, 25, 28, 32 mm and length of 6000-12000 mm in Baku Steel Company LLC—Manufactured according to 2016 and EN 10080-2005 standards.
Pasteur. In Baku Steel Company LLC, the types of square pistachios poured continuously are produced in accordance with GOST380-2005, TŞAZ 3098830-02-2001 with a cross-sectional area of 100x100 mm, 120x120 mm, 125x125 mm, 130x130 mm, and 150x150 mm, and a length of 6000–12000 mm.
Angle. Baku Steel Company LLC manufactures angle shelves with a width of 40x40, 50x50, 63x63, 75x75, 80x80, 100x100, 120x120, 125x125 mm, and a length of 6000-12000 mm according to GOST535-2005, GOST8509-93, and GOST380-2005 standards.
Katanka. Katanka assortments are produced by Baku Steel Company LLC in accordance with GOST30136-95, GOST2590-81, and GOST380-2005 with a diameter of 5.0, 5.5, 6.5, 7.0, 8.0, and 9.0 mm and a weight of 500-750 kg in the form of a bundle.
Ikitavr. In Baku Steel Company LLC, Ikitavr assortment shelves are produced in accordance with GOST535-2005, GOST8239-89, and GOST380-2005 with a height of 100, 120, 140, and 160 mm and a length of 6000-12000 mm. According to the order, the height of the shelf of the two can be increased to 180, 200, 220, 240, 270, and 300 mm.
Channels. Channels are produced at Baku Steel Company LLC in accordance with GOST535-2005, GOST8240-97, and GOST380-2005 standards with a height of 80, 100, 120, 140, and 160 mm and a length of 6000-12000 mm.
A square. Square assortments are produced by Baku Steel Company LLC in accordance with GOST535-2005, GOST 2591-88, and GOST380-2005 with cross-section 10x10, 12x12, 14x14, 16x16, 18x18, 20x20, 22x22, and 24x24 mm, length 6000–12000 mm.
Wand. Rods of Baku Steel Company LLC have diameters of 10, 12, 14, 16, 18, 20, 22, 25, and 28 mm and lengths of 6000-12000 mm in accordance with GOST 535-2005, GOST 2590-2006, and GOST 380-2005..
Round Pasteur. Baku Steel Company LLC produces round pies with a diameter of 130, 160, 170, 190, and 220 mm and a length of 6000–12000 mm in accordance with AZS876-2016 (GOSTP53932-2010) Standards.

Research results and discussion: As a result of the research, it was found that there are some mistakes and errors in the way technology is used here.
Swelling of one or more tongues of a square or rectangular paste may be shown. The side of the paste profile is usually in two of the four languages in the opposite direction. The multiplication of ingot tongues occurs when the total stresses due to ferrostatic pressure and heating of the ingot surface layers exceed the strength of the shell, and the support system in the secondary cooling zone does not ensure its geometric stability. Resistance to foaming is determined by the shape of the ingot, the size of the cross-sectional area, and the strength of the shell.

Rhombicity may be accompanied by longitudinal creases and external and internal cracks. This type of defect develops in rectangular passages (wide tongues). The maximum defect occurs in the flat product (slabs). Inside (narrow) tongues, it can occur as a result of the strong compression of the ingot on the rollers.
Drowning. Swelling deformation in one or more tongues of a square paste: a distortion of the profile by bending the tongue of the paste. Continuous spillage can lead to a swollen surface.
Oval. Ovalness is expressed by distorting the cross-sectional shape of the paste, i.e., by violating the proportions of the cross-section. The size of the oval is the ratio of the difference in deviations from the dimensions of the cross-section of the paste, in %.
Deviation in length. The longitudinal deviation is a change in the size of the sample, both positive and negative.
Curvature (swelling). Curvature (swelling) - is the bending of an assorted paste vertically or horizontally. It may be at one end of the pasta. It occurs in the first or subsequent passages of a series or after accidental stoppages.

Results: -BSC is a leading enterprise in the Republic of Azerbaijan for steel production and currently has an annual production capacity of 1,000,000 tons.
-Two electric arc furnaces and three rolling lines are involved in the implementation of the current production volume.

АНАЛИЗ ПРОЦЕССА РАЗЛИВКИ ПРЯМОУГОЛЬНЫХ КВАДРАТНЫХ ПАСТОВ, ИСПОЛЬЗУЕМЫХ В АРМАТУРНОМ ВАЛКЕ

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РЕЗЮМЕ

В исследовании показаны косвенные способы искривления профилей прямоугольных квадратных проходов, используемых при армировании или изменении размеров отдельных участков их периметра. Было установлено, что если профиль пастеризатора испорчен, даже при отсутствии трещин, это может вызвать ряд проблем на следующих этапах.
Ключевые слова: арматура, прокатка, фисташка, квадрат, трещина, прямоугольник, дефект, процесс заливки.

ARMATUR YAYMADA İSTİFADƏ OLUNAN DÜZBUCAQLI KVADRAT PƏSTAHLARIN TÖKÜLMƏ PROSESİNİN TƏHLİLİ

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XÜLASƏ

İşdə armaturyaymada istifadə olunan düzbucaqlı kvadrat pəstahların profillərinin təhrifi və ya onların perimetrinin ayrı-ayrı sahələrinin ölçülərinin dəyişməsi pəstahların çatlarla zədələnməsinin dolayısı yolları göstərilmişdir. Təyin edilmişdir ki, pəstahların profilinin təhrifi, hətta o çatlarla müşahidə olunmursa da sonrakı emalda bir sıra çətinliklər yaradır.

Açar sözlər: armatur, yayma, pəstah, kvadrat, çat, düzbucaqlı, qüsur, tökülmə prosesi.

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ABOUT THE SYSTEM OF ELECTRIC TRANSMISSION OF INSTALLATIONS OF ELECTRIC SUBMERSIBLE CENTRIFUGAL PUMPS USED IN OIL WELLS

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ABSTRACT

The article details the materials on the elements of the electrical transmission system of electric submersible centrifugal pumps, which are used in flooded oil production wells to lift products in the form of a water-oil mixture from the bottom to the surface. At the beginning of the article, the device and purpose, the principle of operation, the technical characteristics of the cable, the order of descent into the well on the tubing string, the required geological and physical conditions for its normal operation in the well are presented. Then the purpose of using the transformer and autotransformer, the device, purpose, principle of operation and satisfactory conditions for their normal operation are announced. Further, the control station and the main functions performed by it, the purpose of each element of this station, and its operation in cold and moderate climatic conditions are described in detail.
Keywords: Electric submersible centrifugal pump, cable, transformer, autotransformer, control station, submersible motor, tubing, belt, geological and physical conditions.

Topicality of the research: The electrical transmission system of the installation of an electric centrifugal submersible pump (ECPNU) includes: a cable, an autotransformer (or transformer) and a control station [1÷5].
Electric energy is transmitted to the submersible electric motor (SEM-u) by a three-core cable, which is lowered into the wells parallel to the tubing string (tubing). The cable is attached to the outer surface of the tubing string by means of two metal belts in each pipe. The cable operates in very difficult geological and physical conditions. Its upper part is in a gaseous environment, and the lower part is in a high-pressure oil environment. During tripping operations, it is subjected to powerful mechanical stresses, including compression, friction, jamming between the production string and the tubing string, etc. High voltage electrical energy is transmitted with the cable. The use of a high voltage PED makes it possible to reduce the electric current and the diameter of the cable. However, to power the SEM, a high voltage cable must have a more reliable and heavy protective coating. The need to lower the cable next to the ESP requires a reduction in its size in diameter. With this in mind, a flat cable is produced with a thickness two times less than the diameter of a round cable. In ESPNU-key, used cables are available in two types: with round and flat cross-sections. Figure 1 shows the diagrams of these cables. The cables are insulated with an oil resistant rubber or polyethylene coating. Cables with a rubber coating and a circular cross section are designated as KRBK, and cables with a polyethylene coating are encrypted as KPBP.

Flat cable is mainly used in the lower part of the tubing string, where the pumping unit operates; and a round cable descends at the top. In recent years, a wide place has been given to the use of flat cables in all depth intervals (Fig. 1).

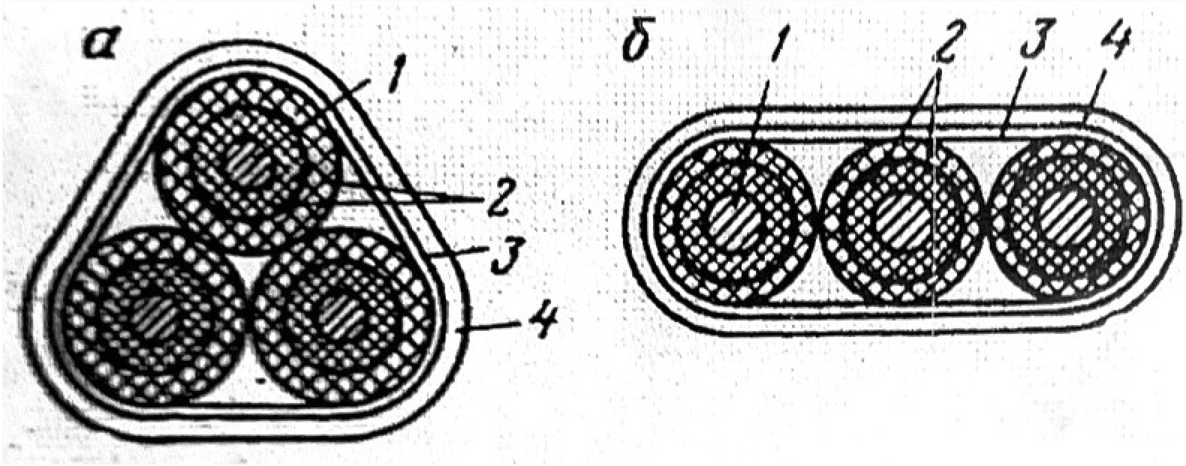


Figure 1. Cable diagrams: 1-core of the cable, 2-electrical insulation and thickness of protection from the external environment, 3-coating under the armor.

Polyethylene-coated cables are 26 ÷ 35% lighter than rubber-coated cables. Rubber-coated cables are intended for use in conditions with a rated voltage of 1100 V, with an ambient temperature of 900 C and a pressure of not more than 1 MPa. From cables with a polyethylene coating, it can be used at a voltage of 2000 V, with a temperature of 1200 C and a pressure of 2 MPa. This cable is very resistant against gas and pressure.
Cables have active and reactive resistances. Active resistance depends on the size of the cross section, and sometimes on temperature. For example, if the cross-sectional area of the cable is 16.25 and 35 mm2, then, respectively, the active resistance is 1.32; 0.81 and 0.6 Ohm/km. And reactance depends on cos φ. With cos φ= 0.86 ÷ 0.9, reactance is approximately 0.1 Ohm/km. There is a loss of electrical power in the cable. This loss is 3 ÷ 15% of the total energy loss. In a cable, power losses are associated with electrical voltage losses. And the voltage loss depends on the current strength, cable temperature, its cross-sectional area, etc., that is, on these factors.
Its calculation is carried out by the formulas of electrical engineering. These losses are typically between 25 V/km and 125 V/km. Therefore, at the wellhead, the voltage supplied to the cable from the nominal value required by the SEM must be greater by the number of losses. Types and technical indicators of cables are presented in table 1.

Table 1. Technical indicators of the control station of the complex installation.

Indicators	ШГС 5804-49А3У1	ШГС 5804-59А3У1	КУПНА 79-29А2У1
Main’s voltage, V	380	380	300
Voltage in the main line (transformer or	2300	2300	-

autotransformer output)			
Current power in the main line (autotransformer or input transformer) A	250	400	-
Current power in the main line (at the output of an autotransformer or transformer) A	74	150	100
Voltage in the control line curve, V	380	380	220
Weight, kg	290 ± 15	335 ±15	1100 ±50

The transformer (or autotransformer) from the network receives the voltage (380V), considering the voltage loss in the cable in the amount of operating voltage required by the PED (470 ÷ 2300 V) in order to increase it is applied. The transformer has the ability to regulate the output voltage to increase and decrease. In cases where the transformation ratio (increase-decrease) does not exceed 2.5. A transformer is used, otherwise an autotransformer. The efficiency factor (COP) is in the range of 98 ÷ 98.5%.
Depending on their power, their weight is 280 ÷ 1240 kg, and the geometric size is from 1060 x 420 x 800 mm to 1550 x 690 x 1200 mm. The types of transformers and their technical indicators are given in tables 3 and 4. In the transformer code, for example, in TMPN 100/3 - 73 XL-e three-phase transformer, M-filled with oil, PN-for oil wells, 100-rated transformer power (KV- A; 3-class high voltage windings, 73-year of manufacture; XL-made according to climatic conditions; in cold climatic conditions XA, for moderate conditions is shown with Y-ohm; 1 shows the category of placement, working in open weather conditions.
The control station of the ESP Unit performs the following functions: manually and automatically put into operation or stop; ensure operation in accordance with the specified installation program; guard; during the operation of the installation, monitor the change in the number of parameters, if they change the operating mode of the pump and carry out hydrodynamic studies with steady inflows. In cable and PED in case of short circuit; in case of overloading of the pump; upon termination of the source of electrical energy in the "cable-motor" system, a sharp decrease in insulation resistance, etc. in such undesirable cases of the regime, the control station automatically leaves the operation of the ESP.
When the power of the SEM-I is in the range from 28 kVt to 100 kVt, then the SHGS 5804 complex of the control station is used, and in cases above 100 kVt, the KUPNA-79 complex of the control station is used. Both integrated control stations are housed in a metal cabinet. The station is intended to be used at weather temperatures from - 350 C to + 400 C. They should not be placed in environments that do not have electrically conductive dust particles, imaginable mixtures of gas and steam, etc. the station must be protected from moisture, external shocks and vibrations. The weight of the control station is 160 kg, the geometric dimensions are 1300 x 850 x 400 mm (Table 1.). Table 1 presents the technical indicators of the control station and the complex installation [6 ÷ 10].

Conclusion: 1. The article details the information about the elements of the electrical transmission system of installations of electric submersible centrifugal pumps used in oil wells.

2. At the beginning, the purpose of the device, the principle of operation, the technical characteristics of the cable, the order of descent into the well on the tubing string, the required geological and physical conditions for its normal operation in the well are given.
3. The purpose of using the transformer, its device, the principle of operation, and the satisfactory conditions for its normal operation are explained.
4. The control station and the main functions performed by it, the purpose of each element of this station, and its operation in cold and moderate climatic conditions are described in detail.

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О СИСТЕМЕ ЭЛЕКТРИЧЕСКОЙ ПЕРЕДАЧИ УСТАНОВОК ЭЛЕКТРИЧЕСКИХ ПОГРУЖНЫХ ЦЕНТРОБЕЖНЫХ НАСОСОВ, ПРИМЕНЯЕМЫХ В НЕФТЯНЫХ ОБВОДИВШИХСЯ СКВАЖИНАХ

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РЕЗЮМЕ

В статье подробно изложены материалы об элементах системы электрической передаячи установок электрических погружных центробежных насосов, которые применяются в обводнившихся нефтяных эксплуатационных скважинах для подъема продукции в виде водо-нефтяной смеси с забоя на дневную поверхность. В начале статьи представены устройство и назначение, принцип действия, техническая характеристика кабеля, порядок спуска в скважину на колонне НКТ, требуемые геологофизические условия для его нормальной работы в скважине. Затем объяенена цель использования трансформатора и автотрансформатора, устройство, назначение, принцип работы и удовлетворительные условия их нормальной работы. Далее подробно опибсывается станция шуправления и основные функции выполняемые ею, назначение каждого элемента этой станции, о ее работе в холодных и умеренных климатических условиях.
Ключевые слова: электропогружной центробежный насос, кабель, трансформатор, автотрансформатор, станция управления, погружной электродвигатель, насосно - компрессорные трубы, пояс, геолого-физические условия.

NEFT QUYULARINDA İSTİFADƏ EDİLƏN MƏRKƏZDƏNQAÇMA ELEKTRİK DALMA NASOSLARIN QURAŞDIRILMASINA ELEKTRİK ÖTÜRÜLMƏSİ SİSTEMİ HAQQINDA

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XÜLASƏ

Məqalədə su-neft qarışığı şəklində məhsulların dibdən səthə çıxarılması üçün su basmış neft hasilat quyularında istifadə olunan elektrik dalğıc mərkəzdənqaçma nasoslarının elektrik ötürücü sisteminin elementləri üzrə materiallar ətraflı şəkildə verilmişdir. Məqalənin əvvəlində qurğu və təyinat, iş prinsipi, kabelin texniki xarakteristikaları, boru kəməri üzrə quyuya enmə qaydası, onun quyuda normal işləməsi üçün tələb olunan geoloji və fiziki şərait göstərilir. Sonra transformator və avtotransformatordan istifadə məqsədi, cihazı, təyinatı, iş prinsipi və onların normal işləməsi üçün qənaətbəxş şərait elan edilir. Daha sonra idarəetmə stansiyası və onun yerinə yetirdiyi əsas funksiyalar, bu stansiyanın hər bir elementinin təyinatı, soyuq və mülayim iqlim şəraitində işləməsi ətraflı təsvir edilmişdir.
Açar sözlər: Elektrikli sualtı mərkəzdənqaçma nasosu, kabel, transformator, avtotransformator, idarəetmə stansiyası, sualtı mühərrik, boru, kəmər, geoloji və fiziki şərait.

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